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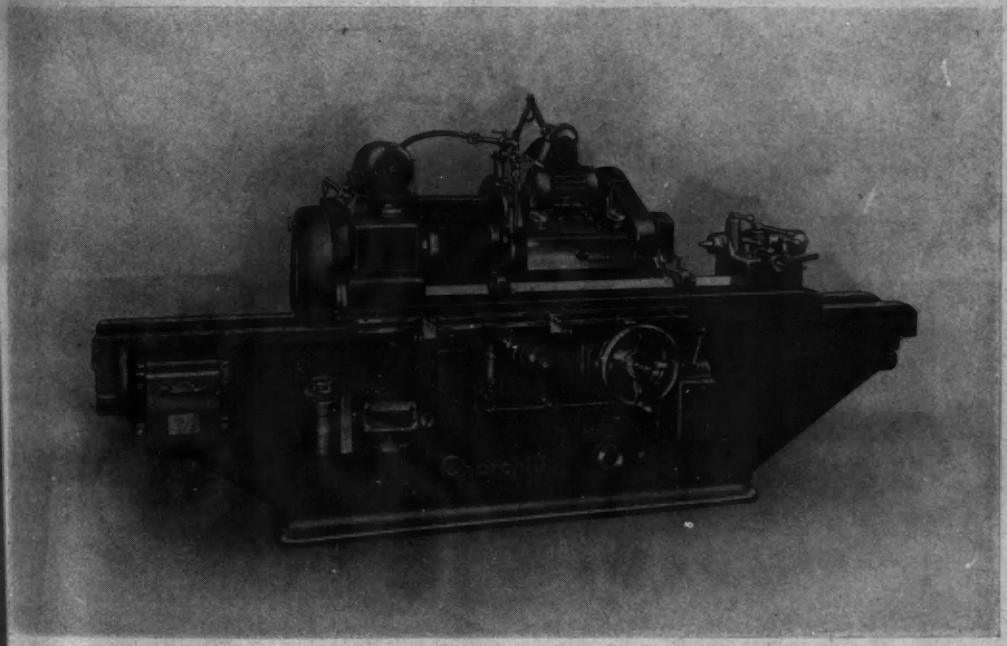

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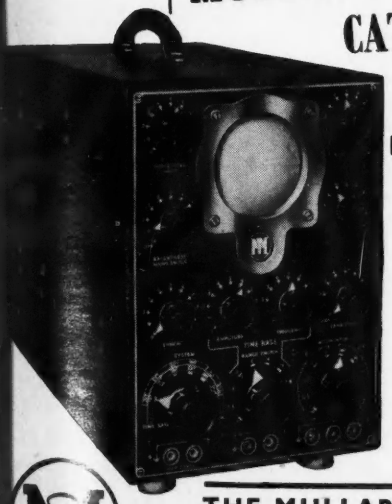

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THE STRENGTH AND SAFETY OF WELDED JOINTS

By M. Ros. (From *Schweizer Archiv*, Vol. 12, Nos. 1, 2, and 3, January, February, and March, 1946, pp. 1-12, 48-63, and 73-92 respectively, 57 illustrations.)

THEORY.

THE permissible stresses in welded pipe lines, boilers and pressure vessels, and in welded steel buildings and bridges were based on series of tests carried out at the Zürich Testing Laboratories (EMPA) from 1934-1945. These tests confirmed the validity of Mohr's theory in its application to units under bi-axial (plane) and tri-axial (spacial) stresses. Up to the stage immediately before failure through shear the elastic as well as the plastic deformations (due mainly or exclusively to sliding) correspond to a nominal stress

$$\sigma_g = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_3\sigma_1}$$

The nominal strain in the plastic range corresponding to the nominal stress is given by

$$\delta_g = \sqrt{\delta_1^2 + \delta_2^2 + \delta_3^2 - \delta_1\delta_2 - \delta_2\delta_3 - \delta_3\delta_1}$$

The letters σ_1, σ_2, and σ_3 represent the stresses in the direction of the principal axes while δ_1, δ_2 and δ_3 indicate the plastic deformations in the same directions. The resultant shear stress, corresponding to the above nominal axial stress σ_g and acting in the octahedron plane of the cube element, is given by

$$\tau_f = \frac{\sqrt{2}}{3} \times \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_3\sigma_1}$$

Similar formulæ are given for spacial stresses and combination of shear stresses and axial stresses. It is shown that the EMPA theory, with sufficient accuracy, can also be applied to fatigue, simply by replacing the static stress limits (e.g. yield point) by the respective fatigue limit.

Extensive tests were devoted to ascertaining the fatigue properties of butt welded joints and of pure weld metal, in comparison with those of the steel plates as received. Fig. 1 explains the results in graph form. Different qualities of steel were used :

STN	of approx.	40	kg/mm²	min.	tensile	stgth	
MI	,, ,,	35–44	,,	—	,,	,,	
MII	,, ,,	41–50	,,	—	,,	,,	
ST44	,, ,,	44	,,	min.	,,	,,	
STHW	,, ,,	52 or 54	,,	,,	,,	,,	

The graph refers to the upper fatigue values obtained from one million repetitions of load with zero as the lower limit (Ursprungsfestigkeit). Test results are included for specimens with butt welds running at 45 deg.

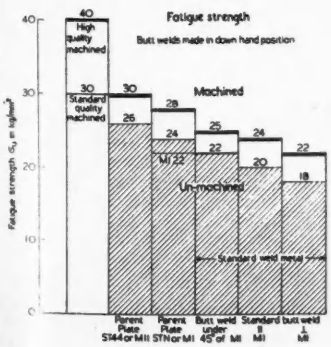


Fig. 1. Fatigue limits for weld metal, parent plates and butt welds of various types and materials in machined and un-machined condition.

towards the direction of the tensile force (" spiral " seams) ; if machined their endurance limit practically equals that of the plain plate " as received ", and is well beyond that of the right angle butt weld in unmachined condition. Fatigue failure of the spiral butt welded specimen, if machined and practically non-porous, always takes place outside the weld in the plain plate, beyond its machined portion (see Fig. 2), because of the

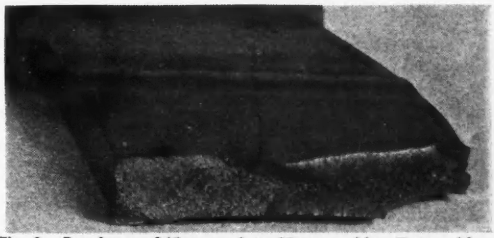


Fig. 2. **Specimen of 15 mm. plate, 65 mm. wide. Butt weld under 45 deg. to the direction of pull, machined. Fatigue limit 25·4 kg/mm², fracture through un-machined plain plate, well away from the weld.**

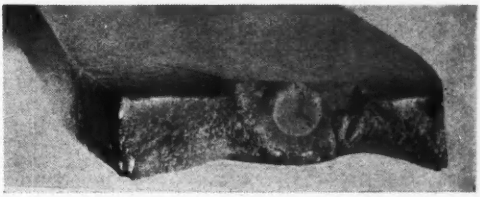


Fig. 3. **Butt weld under 45 deg., machined. Fatigue limit 26·1 kg/mm², fracture right across the plate, initiated by a pinhole in the weld.**

15 to 20 per cent higher fatigue strength of the plate in the machined zone compared with that of the unmachined zone. Any small pinholes in the machined weld may, however, easily cause fatigue failure through the weld, as shown in Fig. 3. Although the initial crack obviously started at the pinhole in the spiral butt weld the fracture did not follow the weld but went right across the specimen.

The type of failure of a 45 mm. single V-joint, with back sealing run, machined, under static load, is shown in Fig. 4.

In laying down a set of permissible working stresses, the following relations were used :

For boilers, pressure vessels, wagon tanks and

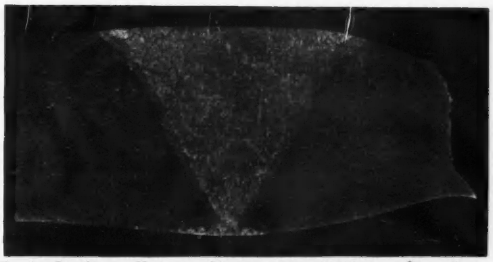


Fig. 4. **Static tensile test. 45 mm. plate of 36 kg/mm² tensile strength, with single V-joint, back sealing run, machined, tensile strength in weld 41 kg/mm². Lateral contraction causes a 14 per cent. increase in unit strength.**

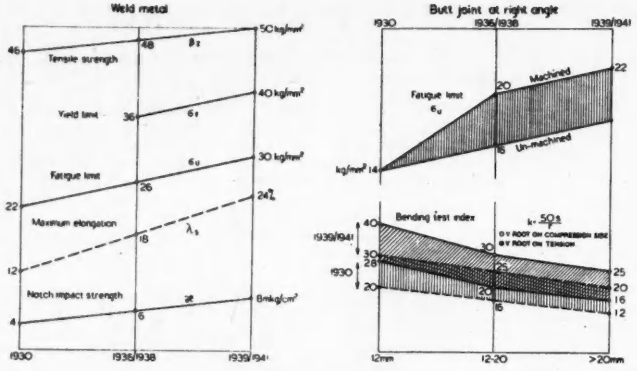


Fig. 5. **Weld metal and butt welded joints, progress in mechanical properties since 1930.**

pressure pipe lines :

$$\alpha_B = \frac{\text{permissible stress in welded joint}}{\text{permissible stress in plate}};$$

for welded steel structures, particularly bridges :

$$\alpha_N = \frac{\text{permissible stress in welded joint}}{\text{permissible stress in riveted joint}}.$$

The progress made since 1930 up to 1945 with regard to the mechanical properties of the butt welded joint is illustrated in Fig. 5. The increases are in the region of 30 per cent for the tensile endurance strength, and of 60 per cent for the bending test index, both with the root of the V-weld on the compression side or on the tension side. These remarkable improvements are closely connected with the development of high quality covered electrodes, the purity, uniformity and centricity of the core wire, and with simultaneous progress in developing the metallurgical and the mechanical testing procedures. The hardness of the deposited weld metal has been kept down to a minimum, while its plasticity was considerably increased both at normal temperature and at temperature variations down to —60 deg. C. and up to + 700 deg. C. Its endurance strength and its izod values were improved and its sensitivity against ageing and frost practically eliminated. A remarkable homogeneity has been achieved in the structure of the weld metal and in its static and dynamic properties, indicated by ductile behaviour under load and a weld surface of low notch sensitivity.

BUTT WELDS IN PRESSURE PIPE LINES, BOILERS AND PRESSURE VESSELS.

The advantages of the spiral butt welds (their optimum inclination is 45 deg. to the longitudinal axis) are theoretically less for the higher quality welded joints (Class I) than for Class II, both in machined and un-machined condition, as compared with the orthodox circumferential butt weld. In practice, however, tests with spiral welded specimens show higher values of fatigue strength. Slightly higher values of permissible stress for spiral welds were therefore adopted. There is such good agreement with the new EMPA formula of 1945 that circumferential and spiral welds can be combined without hesitation.

The extensive series of tests included fatigue tests with specimens V– and X–butt welded at right angles, parallel and at varying angles to the direction of stress, and tests to destruction of vessels by inner pressure, with detailed measurements of stresses and deformations. Factors of safety are indicated in some of the numerous graphs, assuming failure

(*a*) through yield of the parent plate
(*b*) through fatigue of the welded joint
(*c*) through destruction under static inner pressure.

Against fatigue failure the required factors of safety are at least 1·6 for pressure pipe lines and wagon tanks, 1·7 for distribution pipes, and 2·0 for steam boilers and pressure vessels (see Fig. 6). All these factors refer to the inner pressure in service (not in acceptance tests) and to plates of not more than 60 mm. thickness. Special precautions have to be taken when thicker plates are to be welded, covering sequence of work, electrode diameter for individual runs, preheating etc.

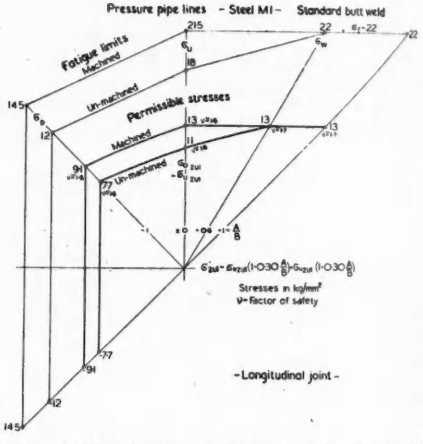


Fig. 6. **Permissible ring stresses in pressure pipe lines, butt welded, steel MI, welds machined, un-machined and stress relieved by annealing.**

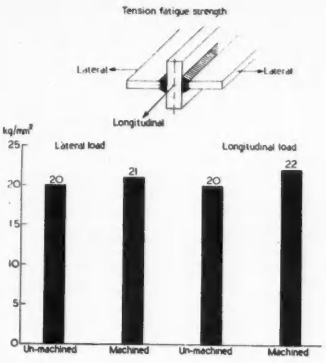
FILLET WELDS AND BUTT WELDS IN STEEL CONSTRUCTION.

Side fillet and end fillet joints were amply investigated, the latter also with the cruciform type of specimens, under static and fatigue loads. Graphs and tables give all the required α — values, as the ratio of permissible stress for the welded joint to permissible stress for the parent plate (and also to permissible stress for the riveted joint). The machining of fillet welds was shown to result only in a slight improvement of the fatigue strength at right angle to the weld. What is of decisive importance, however, is the sound and thorough penetration into the root of the fillet weld.

Another series of graphs provides sets of permissible stresses in tension, compression and shear, parallel and at right angle to the welds, to be used for welded construction of buildings, road and railway bridges, hydraulic structures, cranes, conveyor plants and rolling stock. Factors of safety against fatigue are also indicated.

The K-type joint as used in the cruciform specimen of Fig. 7, ranges between an end fillet and a butt joint. In view of the greater care required in preparation and welding, the permissible stresses were fixed 10 per cent lower than those for butt welds.

The permissible stresses are throughout dependent on the ratio A/B, e.g. the proportion of the minimum to the



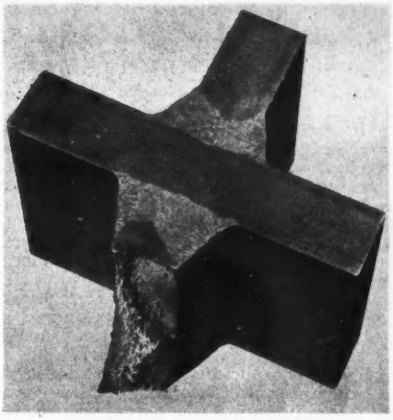


Fig. 7. **K-type joint, machined. Fatigue limit under lateral pull 21 kg/mm². Fracture started at point of undercutting. The graph on the left shows nearly equal values of fatigue strength under lateral and longitudinal loads. 5 per cent and 10 per cent increases are obtained by machining.**

maximum value of repetition load for a particular joint. Generally the permissible axial stress is given by :

$$\sigma = \sigma_0 \left((1 + 0{\cdot}3 \frac{A}{B} \right)$$

and the permissible shear stress similarly by :

$$\tau = \tau_0 \left(1 + 0{\cdot}3 \frac{A}{B} \right).$$

The coefficient 0·3 applies to steel STN (of 40 kg/mm² minimum tensile strength) while 0·4 is used in its place in otherwise similar formulæ for STHW of higher tensile strength.

σ_0 and τ_0 are the permissible stresses (axial or shear) of the parent plate under fatigue loads, with zero as limit. This is clear when assuming $A = 0$; then $\sigma = \sigma_0$ and $\tau = \tau_0$. With $A = B$, as in statically loaded structures, like buildings, $\sigma = 1{\cdot}3\ \sigma_0$ and $\tau = 1{\cdot}3\ \tau_0$.

SUMMARY AND CONCLUSIONS.

The value of welded joints in theory and design is expressed by coefficients α ; generally :

$$\alpha = \frac{\text{fatigue strength of welded joint}}{\text{fatigue strength of plate}}.$$

The fatigue strength figures refer to one million repetitions of load, with zero as the lower limit (in tension) or as the upper limit (in compression) ; this is the meaning of " Ursprungsfestigkeit ". In plate construction, e.g. for boilers, pressure vessels, wagon tanks, pressure pipe lines, etc., the α —value should relate to the parent plate, while in sectional steel construction, e.g. buildings and bridges, the fatigue value for the denominator preferably refers to the riveted joint.

The average fatigue strength of a first class butt-welded joint, un-machined, at right angle to the joint, welded in down-hand position, is about 18 kg/mm², machined 22 kg/mm², with —10 per cent tolerance. With the load parallel to the joint (as in the case of the circumferential seam of a cylindrical tube under internal pressure) the fatigue values are 10 per cent higher.

It is clearly preferable to take the fatigue strength as the basis when deciding upon the value of a butt welded joint. Any faults of the material, or in design, preparation and welding procedure will show their effects much more markedly than under static loads. It is a well-known fact that two butt welded specimens may show a marked difference in their fatigue limits although their static tensile values are practically the same.

The butt weld at an angle (spiral joint) is definitely less sensitive against the welding faults which are so difficult to avoid in practice., e.g. porosity, slag inclusions, undercutting, rough surface of weld and arc craters. The spiral weld therefore deserves preference. This applies particularly to pressure pipe lines, penstocks and vessels under high internal pressure, with large diameters and walls up to 60 mm. thickness, but it is also recommended for bridge construction.

Some of the advantages are indicated by the following percentage figures, all referring to butt welds :

	per cent
Stress relieved by annealing, as against untreated	+ 10
Machined, as against un-machined	+ 20
Spiral weld, as against longitudinal weld, both un-machined	+ 14
Spiral weld machined, as against spiral weld un-machined	+ 14
Spiral weld machined, as against longitudinal weld machined	+ 8

The progress achieved over the last 12 years or so is effectively illustrated by Fig. 8, which refers to the permissible stress in circumferential joints of welded pressure pipe lines, boilers, pressure vessels and wagon tanks.

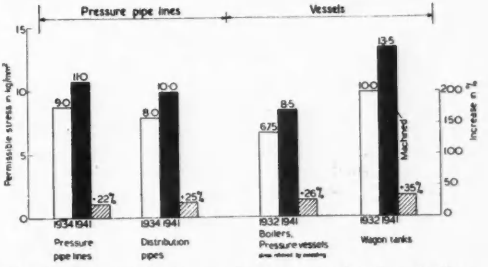


Fig. 8. **Permissible stresses in pressure pipe lines, boilers, pressure vessels, and wagon tanks. Increase in permissible ring stresses since 1932, with plates of MI quality.**

Still more remarkable increases of permissible stress have taken place in welded construction for buildings and bridges, amounting to 25 to 40 per cent. For spiral seams, stress relieved by annealing, 45 to 70 per cent have been gained, and 40 per cent with similar seams in steel structures.

FRANCE

AN ARRANGEMENT FOR SUPPRESSING OSCILLATIONS IN TRANSFORMERS SUPPLYING SELF-RECTIFYING X-RAY TUBES.

By L. MAURICE. (From *Bulletin de la Société Francaise des Électriciens*, No. 57, March, 1946, pp. 98-102, 5 illustrations.)

STUDY of the transformers feeding X-ray tubes, by means of the oscillograph, has shown that they are often subject to oscillations whose frequency in agreement with the circuit constants (secondary capacity, leakage inductance, etc.) is of the order of 1000 to 1500 c/s for the transformers most commonly used in radiology. They arise when the circuit is made, or as a result of instability of the tube, and, having amplitudes reaching double the working voltage, may damage both tube and transformer. To damp them out as quickly as possible a choke, shunted by a resistance used to be inserted in the primary or secondary, but a rectifier shunted by a resistance is now used instead, and experience has shown it to be particularly effective. The arrangement is not new but has wider applications, which are investigated.

The working of a transformer supplying a self-rectifying X-ray tube is first considered in detail. If the transformer is connected to the mains then the voltage applied to the primary is sinusoidal, and the secondary voltage is also sinusoidal to a first approximation, apart from the voltage drop. The secondary current in a saturated X-ray tube is a complex function of the voltage. Theoretically, during a half-cycle the current should grow with the voltage, remain constant when saturation voltage is reached and then decrease, but in practice the tubes are only partially saturated. The more intense the current traversing a tube, the less saturated it is, with the result that the current always has a form intermediate between the theoretical and the sinusoidal.

At the maximum intensity for which the tube is designed the secondary current is very nearly a half sine wave, in phase with the secondary voltage (Fig. 1a). The current is unidirectional, for no current traverses the self-rectifying tube on the second half-cycle, and so the mean secondary current is not zero.

The primary current at any instant is the sum of the secondary current with its sign changed, and the magnetizing current. (To simplify matters a transformation ratio of l is assumed so that " current " may be used, instead of " ampere turns "). Nothing can be said, *a priori*, of its form, but an ammeter connected in circuit does not move, so the primary current is alternating with zero mean value (Fig. 1b).

The magnetic field of the core is the sum of the fields

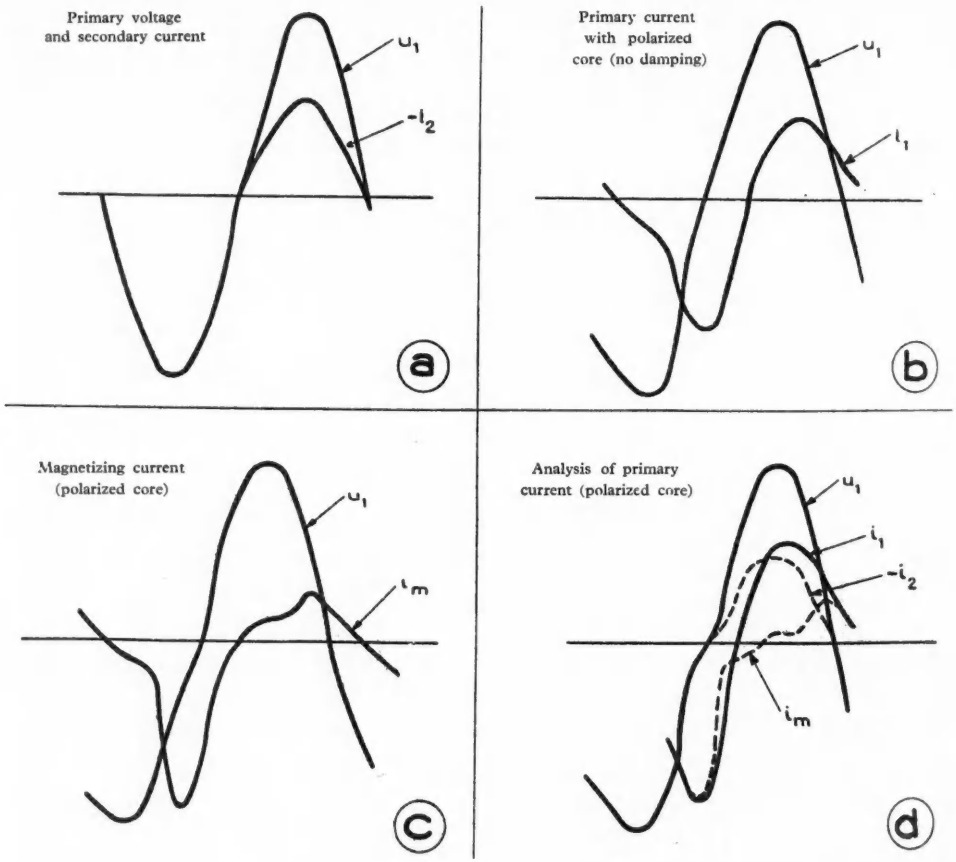


Fig. 1

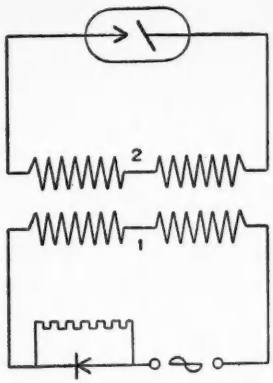


Fig. 2

induced by the primary and secondary. The primary current being alternating, with zero mean value, the mean primary field is zero. The secondary current having a continuous component, the secondary field likewise has a continuous component, which appears in the sum. The core is therefore polarized, and the shape of the primary current is easily explained. On the curve of magnetization of the core, H^c, the polarization of the core corresponds to a continuous flux Φ^c. To this is added an alternating flux $\pm \Phi^a$ due to the sinusoidal primary voltage. The flux thus varies between $\Phi^c - \Phi^a$ and $\Phi^c + \Phi^a$ and the field from $H -$ to $H +$ and the magnetizing current is very asymmetrical. Its form is shown in the oscillogram (Fig. 1c). How this curve is obtained is shown later. Fig. 1d shows that the primary current (Fig. 1b) is actually the sum of the secondary current with sign changed and the magnetizing current (Fig. 1c).

It is emphasized that the polarization of the core depends directly on the secondary load. On open circuit, there is no polarization, and the magnetizing current has the normal shape (Fig. 3a). The oscillogram in Fig. 1b corresponds to a moderate load, and even so shows a definite peak in the magnetizing current, but for heavier loads this peak may attain three or four times the maximum intensity of the direct wave. Since this peak obviously depends on the maximum flux density, on open circuit, in the core, transformers of this type should not be designed with too high flux densities in the first place; for not only do excessive currents result, but there are also heavy iron losses.

It is thus important to depolarize the core, the obvious

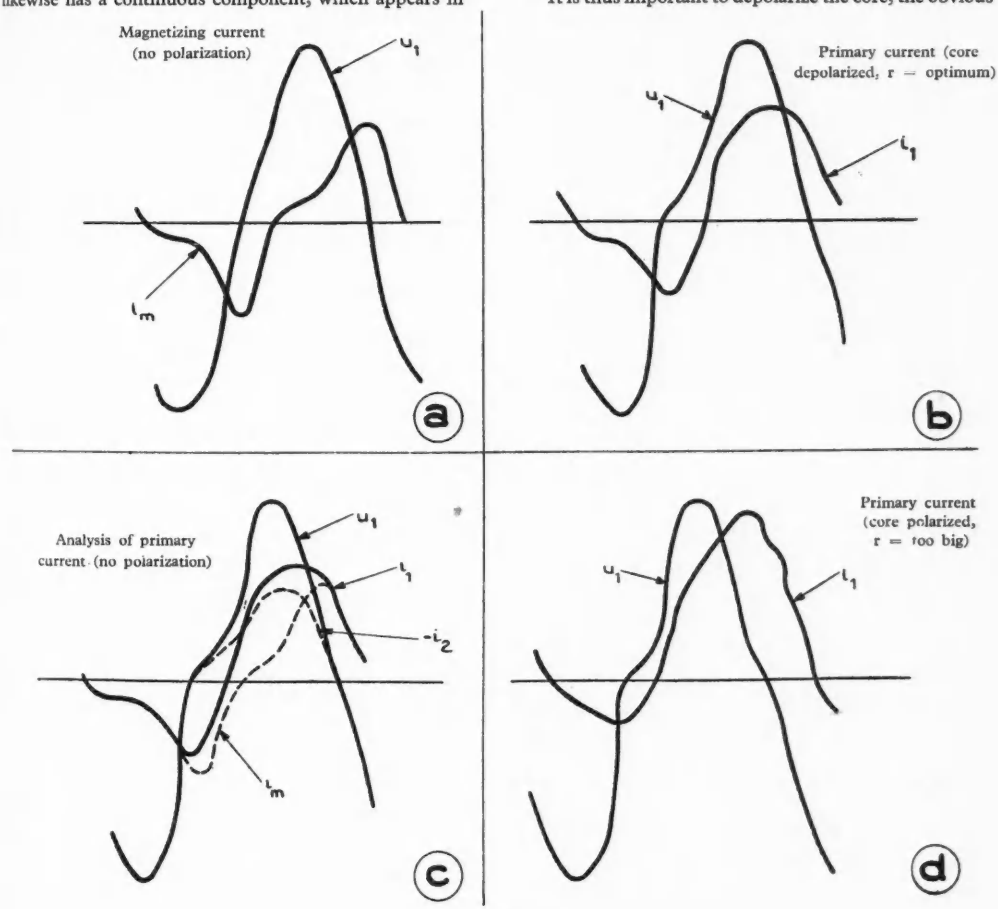


Fig. 3

way being to use a third winding connected to a rectifier and resistor in series, and working in the opposite sense to the X-ray tube. This however doubles the load on the transformer, so that it is better to produce an asymmetry in the primary itself, for which the resistance and rectifier in parallel can be used (Fig. 2). During the direct wave the rectifier allows the current to pass, but on the inverse wave it blocks the way, the current is forced to pass *via* the resistance, and a voltage drop results. The primary current thus becomes asymmetrical and has a continuous component. To obtain the desired result the rectifier must be connected so that this continuous component opposes that due to the secondary.

The size of the component depends on the resistance shunting the rectifier so that for a given load there is an optimum value of resistance for which an exact balance holds and the core is completely depolarized. For smaller resistances there will be a preponderance of the secondary component with incomplete depolarization, while for greater resistances the core will be polarized in the opposite direction, owing to the greater primary component.

There are three methods of finding the optimum resistance :

(1) By insertion of a square-law ammeter in the primary and adjustment of the resistance until the meter indicates a current giving the same number of ampere-turns as in the secondary.
(2) By insertion of a dynamometer-type ammeter in the primary. When the core is depolarized the magnetizing current, and hence the primary current is at a minimum.
(3) By insertion of a wattmeter in the primary. A depolarized core gives minimum copper losses, and hence minimum primary losses.

Oscillograms confirm this theory. Fig. 3b shows the primary current. When the shunted rectifier is fitted and the resistance adjusted by one of these methods, it confirms that the core is completely depolarized. Fig. 3c shows that the primary current is precisely the sum of the secondary current with sign changed and the magnetizing current on open circuit and without damping device. (3a).

The optimum resistance value is a function of the secondary current. If the load decreases the resistance is too great and over-compensates. When the secondary is on open circuit the primary polarization acts alone, and being equal to the secondary polarization, the circuit is in the same state as when the transformer is on load and the damping device not fitted. The magnetizing currents are identical, and an oscillogram can be obtained. Fig. 1c was obtained in this way, and the construction of 1d confirms the theory.

The device of the shunted rectifier in the primary has often been claimed to reduce the inverse wave (the direct wave is that in which current passes through the X-ray tube, the inverse that during which no current passes) ; its action could simply be interpreted thus : during the direct wave the current passes *via* the rectifier, during the inverse wave the current must pass by the resistance creating a voltage drop which limits the amplitude of this wave. This however is not quite correct, because the fact that the inverse voltage does not drop, even when the rectifier is removed, is due, not to the absence of primary current during the inverse wave, but to the current (during this half-cycle a purely magnetizing current) being 90 deg. out of phase with the voltage, apart from hysteresis lag. The maximum voltage is only slightly affected by the voltage drop, the current being almost zero at that moment and it is the foot of the curve which is distorted. With the direct wave, current and voltage are in phase, and there is a considerable voltage drop.

The phenomenon is of the same order when the rectifier is in circuit. The primary current, during the inverse wave, remains a magnetizing current, and the effect of the voltage drop on the maximum inverse voltage is very much less than the simplified explanation would suggest. In fact, however large the shunting resistance, the inverse voltage is always greater than the direct. Nevertheless the effect is not negligible.

SUMMARY.

A rectifier shunted by a resistance connected in the transformer primary has three useful properties.

(1) It damps out oscillations got up when the tube is put in circuit or as a result of tube instability.
(2) Its depolarizing effect on the core reduces transformer losses and permits the use of higher flux densities.
(3) It reduces slightly the inverse wave for a given direct wave.

U. S. A.

STANDARDIZED STEAM TURBINES.

(From *Mechanical Engineering*, Vol. 68, No. 4, April, 1946, pp. 360-361).

THE first standardized steam turbine, a 30,000 kW. machine for electric power generation is in production at the South Philadelphia plant of the Westinghouse Electric Corporation. The unit is the result of a long-felt need to remove the " custom-made " label from such machines.

This development was brought about by a joint committee organized by The American Society of Mechanical Engineers and the American Institute of Electrical Engineers. The A.S.M.E. committee on Standardization of Steam Turbines, under the chairmanship of K. M. Irwin, member A.S.M.E., was appointed in February, 1943. A draft proposal was sent to a large group of manufacturers and users of steam turbines for criticism and comment. The replies received were reviewed, and it was recommended that copies of the proposed standard be sent to British and Canadian interests for their information. This meeting was attended by M. S. Oldacre in his capacity as chairman of the Subcommittee on Standardization of Generators which had been appointed by the A.I.E.E. at its 1943 summer convention. The Joint A.I.E.E.-A.S.M.E. Committee on Steam Turbine-Generators was formed in 1945.

The six sizes of turbines decided upon by this committee as being within the range of the most common of the larger power-plant sizes, and therefore most suitable for standardization, have been defined to condensing units for operation at 3600 r.p.m., between the ratings of 11,500 and 60,000 kW. This is sufficient range to provide power for cities from 10,000 to 50,000 population.

Standardization of turbine sizes will result in reduced prices and shorter deliveries than for non-standard machines, and should permit a decrease in time required by manufacturers to submit performance estimates. The standardization program will in no way hinder production of any size steam turbine-generator needed for purposes for which the standard specifications might not be suited. Further, the standard is not intended to prevent the purchase of duplicates of machines already installed, and for which designs and patterns are available.

Research, too, will be stimulated by this development owing to the amount of effort that can be diverted from custom designing to research.

EXACT MEASURING OF MICRO-INTERVALS BY MEANS OF CATHODE RAY TUBES.

By W. AMREIN and A. v. WATTENWYL (From *Schweizer Archiv*, Vol. 11, No. 9, September, 1945, pp. 257-262, 11 illustrations).

THE main advantage of cathode ray tubes lies in the small inertia of the cathode rays and in the simple way they can be influenced. In small commercial cathode ray tubes operating with anode voltages of 1000 to 2000 the inertia of the cathode ray beam is noticeable only at frequencies above 10^8 cycles per second. In the following it is assumed that for the apparatus in question only this commercial sealed-off glass cathode ray tube is used, and not the high voltage tubes requiring pumps for maintaining a high vacuum The use of commercial type cathode ray tubes entails, of course, a number of disadvantages ; the principal one is the comparative inaccuracy of the oscillogram, due partly to the properties of the tube itself and partly to inaccuracies in the timing circuits, and to electric and magnetic stray fields. Therefore, it must be reckoned that oscillograms taken with tubes of this type show an inaccuracy of about 5 per cent. or more. This, however, does not necessarily preclude their use for precision measurements ; by using appropriate circuits and by taking into account the basic causes of the inaccuracies of the tube, the latter can be used to advantage also for precision measurements.

It is well known that the beam of a cathode ray oscillograph can be given a rotary deflection by applying two sinusoidal potentials with a phase difference of $\frac{\pi}{2}$ – to two pairs of deflecting electrodes spaced at right angles to each other and symmetrical about the undeflected path of the beam. An ordinary circuit diagram for such an arrangement is shown in Fig. 1, which also contains all the elements necessary for the use of the cathode ray tube as a measuring instrument for micro-intervals. The principle of this method for measuring the interval between two short electric impulses is as follows : By applying two positive potential impulses to the terminals 17 which are connected to the anode 14, the luminosity of the trace is increased and the picture shown in Fig. 2, obtained. It can be recorded photographically. The angle α between the starting points of the two impulses 18 and 18′ respectively is proportional to the time interval between the two impulses. If the rotational speed of the cathode ray beam is known, the actual duration of the interval between the two impulses can be established. Modern technique using tuning fork or quartz crystal oscillators as a time base makes it possible to determine the frequency with an error far below 1 per mil. This method of recording requires the complete separation of the traces of the two impulses. To achieve this, the following arrangement is used. The cathode ray tube is fitted with an accelerating electrode 19 to which a potential of several thousand volts is applied. This causes the electrons to impinge on the screen with a higher speed, thus producing a higher luminosity, and at the same time reduces the deflectability of the beam. Other operating conditions being constant, the diameter of the circular

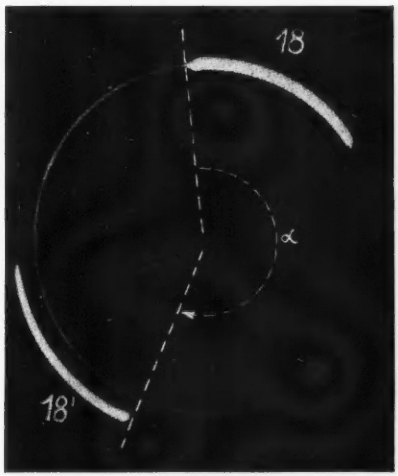
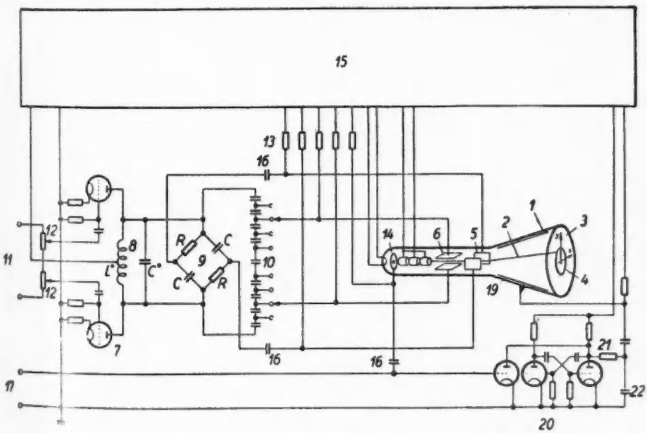


Fig. 2. **Oscillogram of the micro-interval meter.**



1. cathode ray tube
2. cathode ray
3. fluorescent screen
4. circular trace
5. time plates
6. time plates
7. amplifier
8. oscillator circuit
9. phase shifting bridge
10. capacitive potentiometer
11. terminals for time base
12. potentiometer
13. resistances
14. anode
15. mains pack
16. condensers
17. terminals for the impulses to be measured
19. accelerating electrode
20. trigger circuit
21. } circuit elements to provide
22. } delayed action.

Fig. 1. **Circuit diagram of the micro-interval meter.**

trace on the screen is, therefore, reduced if the voltage on electrode 19 is increased. This provides a means of varying the diameter of the trace without deteriorating the focussing of the beam. Now the first impulse is made to reduce the voltage on electrode 19 by about 1000 volts through a trigger circuit 20; the latter is designed for delayed action to allow the first impulse to be recorded before the change of voltage takes place. The reduced voltage on electrode 19 increases the deflectability of the electron beam and thereby increases the diameter of the circular trace on the screen. When the second impulse occurs it is registered as an arc of larger radius which cannot, therefore, overlap with the recording of the first impulse.

Since it is necessary for the accurate measurement of angle a, to record either the centre of the circle or the complete circle itself, it is of advantage to adjust the basic luminosity of the circle so as to produce a faint photographic reproduction on which the impulses appear as bright superpositions. Fig. 3 shows a micro-

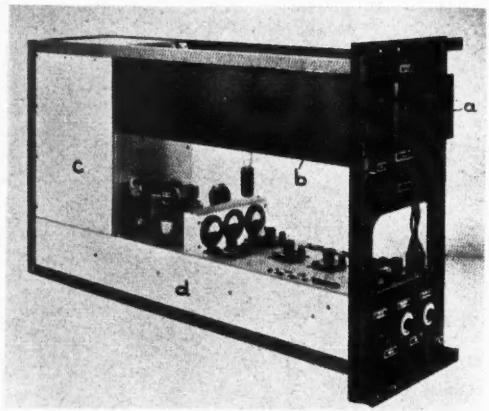


Fig. 3. Micro-interval meter.

(a) Holding frame for photographic film
(b) Casing for cathode ray tube
(c) Casing for oscillator coil and phase shifting bridge
(d) Chassis for amplifier and mains pack

interval meter with an absolute accuracy of two millionths of a second. The time base used is 10000 cycles/sec.

The accuracy of the impulse recording depends on a number of sources of error which are investigated in the following. Fig. 4 shows schematically the two pairs of time plates which deflect the cathode ray beam in the directions x and y proportional to the voltages U_x and U_y. The beam passing through point P would rotate with the constant angular velocity ω provided no sources of error existed. In practice, however, it diverges from the exact position by the angle Δa.

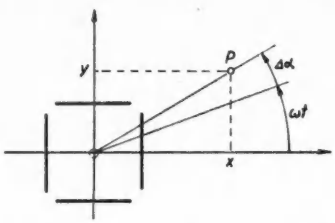


Fig. 4. Time plates and deflection of cathode ray beam.

This is due to :—

(1) Deviation of the phase angle between the potentials applied to the two pairs of time plates by an angle $\Delta\varphi$ from the required value of $\pi/2$.

(2) Inaccuracy in the mechanical mounting of the time plates resulting in their axes not being perpendicular to each other. This error is equivalent to an error $\Delta\varphi$ of the electric phase angle and can therefore be added to the error described under 1.

(3) The amplitude of the deflections caused by one pair of time plates is not equal to that caused by the other. Here it must be taken into account that the amplitude-error can be caused by amplitude-errors of the basic harmonic of the applied voltage U (with the angular velocity ω) as well as by undesirable higher harmonics. The relative error due to the basic harmonic is called ϵ, and the corresponding error in the directions x and y due to the higher harmonics is called $h_x(t)$ and $h_y(t)$.

Other sources of error can be neglected. The deflections x and y in the directions x and y respectively can be analyzed mathematically. For the sake of simplification, the amplitude of the deflection in direction x is assumed to equal 1. Thus

$$x = \cos \omega t + h_x(t) \quad .. \quad .. \quad .. \quad (1)$$

$$y = (1 + \epsilon) \sin(\omega t + \Delta\varphi) + h_y(t) \quad .. \quad (2)$$

Consequently, the angle between the beam through P and, for instance, the abscissa x is :

$$\omega t + \Delta\varphi = \tan^{-1} \frac{(1 + \epsilon) \sin(\omega t + \Delta\varphi) + h_y(t)}{\cos \omega t + h_x(t)} \quad (3)$$

From this equation, by separating the small quantities Δa, ϵ and h, it follows :

$$\Delta a = \Delta\varphi \cos^2 \omega t + \frac{\epsilon}{2} \sin 2\omega t + h_y(t) \cos \omega t - h_x(t) \sin \omega t \quad (4)$$

While $\Delta\varphi$ and ϵ are constant for a given apparatus, the values for $h_y(t)$ and $h_x(t)$ are determined by the higher harmonics and are more or less involved functions of time which vary from case to case.

For the sake of simplification, from now on only the relationship between Δa, ϵ and $\Delta\varphi$ will be examined. Thus it is possible to put

$$\Delta a = \Delta\varphi \cos^2 \omega t + \frac{\epsilon}{2} \sin 2\,\omega t \; ; \; .. \quad .. \quad (5)$$

Δa can assume two extreme values, namely

$$\Delta a_{max} = \frac{1}{2}\Delta\varphi \pm \frac{1}{2}\sqrt{\epsilon^2 + \Delta\varphi^2} \quad .. \quad (6)$$

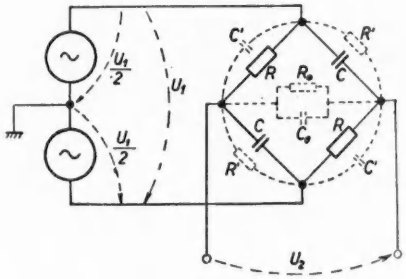


Fig. 5. Phase shifting bridge.

This represents the maximum angular error for one impulse. The maximum angular error between 2 impulses can, therefore, not exceed

$$\Delta\alpha_{\text{total}} = \Delta\alpha_{1\,\max} - \Delta\alpha_{2\,\max} = \pm\sqrt{\epsilon^2 + \Delta\varphi^2} \quad (7)$$

To ensure, for instance, an accuracy of

$$\Delta\alpha \leqq 4\pi/100 = 0{\cdot}1,$$

ϵ and $\Delta\varphi$ must not exceed a few per cent. While it is relatively simple to keep the amplitude error ϵ sufficiently small by adjusting the voltage on one pair of time plates by means of a potentiometer, conditions are more complicated in the case of the error $\Delta\varphi$ of the phase angle. Quite often the mounting of the time plates in commercial cathode ray tubes shows an error of 2 to 3 degrees or about 3 to 4 per cent. To this error has to be added the error of the phase angle between the voltages applied to the time plates, caused by unwanted capacities and resistances parallel to the branches of the phase-shifting bridge 9 in Fig. 1. Fig. 5 shows the same bridge with the unwanted capacities and resistances added; it has been assumed that the whole arrangement is symmetrical. To obtain a phase angle of $\frac{\pi}{2}$ between the voltages U_1 and U_2, the following condition must be fulfilled :

$$\omega RC \geqq 1 + \frac{R}{R_0} - \frac{C_0}{C} \quad .. \quad .. \quad (8)$$

assuming that

$$R \ll R' \; ; \; R \ll R_0 \; ; \; C' \ll C \; ; \; C_0 \ll C. \; .. \quad (9)$$

Under the same assumption, the calculated difference between the amplitudes of U_2 and U_1 will be

$$-\frac{R}{R'} - \frac{C'}{C} - \frac{R}{R_0} - \frac{C_0}{C} \quad .. \quad .. \quad (10)$$

and the calculated angle $\Delta\varphi$ between U_2 and the direction perpendicular to U_1 will be

$$\Delta\varphi = \frac{R}{R_0} - \frac{C_0}{C} - \frac{\Delta\omega}{\omega} - \frac{\Delta R}{R} - \frac{\Delta C}{C} \quad (11)$$

In discussing this equation, the small value of R/R_0 can be assumed constant and therefore of no interest. $\Delta\omega/\omega$ can be sufficiently reduced to make it negligible. $\Delta R/R$ and $\Delta C/C$ are the relative variations of the main elements of the bridge, caused mainly by changes of temperature, and can only be kept within reasonable limits by a proper choice of the constituent parts of the bridge. Only C_0/C remains. The unwanted capacity C_0 is due to the time plates of the cathode ray tube and to the leads ; it constitutes the most uncertain element in the whole discussion. It may change by as much as 5pF merely when the cathode ray tube is changed. If the angular error caused by the stray capacity is required to remain below a certain limit, say 0·5 per cent, a corresponding lower limit is set for the main capacity C of the bridge. The maximum stray capacities of the tube determine, therefore, the main capacity C of the phase shifter circuit.

Finally, attention must be paid also to the higher harmonics of the time base, since, according to equation (4), their full value appears as a component of the angular error. They must be kept as small as possible,

Fig 6. **Equivalent load of oscillator circuit.**

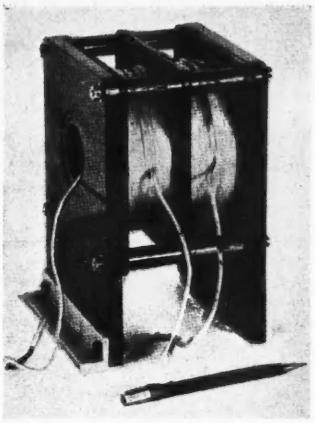


Fig. 7. **Oscillator coil.**

which can best be achieved by a high "quality factor" $Q = R : \sqrt{L/C}$ of the oscillator circuit L^* C^*. This circuit, however, carries the bridge 9 as a load, which in this case is equivalent to a single resistance R with a capacity C connected in parallel to it (see Fig. 6). For instance, if R is 10000 Ω and a "quality" of 100 is required, $\sqrt{L/C}$ will only be 100 Ω. However, the actual quality obtained will be somewhat smaller due to the losses in coil L^*, and due to the fact that the amplifier valves also constitute a load on the oscillator circuit.

The oscillator coil L^* of 4 Millihenry shown in Fig. 7 was wound of stranded wire, has no iron core and the quality attained was 280 at 10000 cycles/sec. This made it possible to keep the quality of the oscillator circuit above 70, which is adequate to suppress sufficiently the higher harmonics, provided the amplifier valves operate within the linear part of their characteristics.

The accuracy of this micro-interval meter can be tested for instance by applying to the anode impulses of a frequency which is a multiple of the frequency applied to the time plate, as shown in Fig. 8.

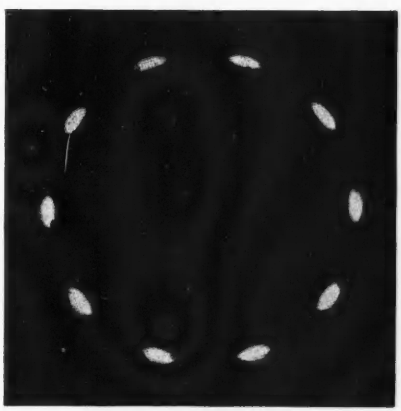


Fig. 8. **Experimental control of accuracy of angle.**

RUSSIA

HEATING AND SURFACE HARDENING OF TOOLS BY HIGH-FREQUENCY INDUCTION.

According to tests conducted at The Gorki Automobile Works. (From *Promishlennaya Energetica*, No. 3, 1946, pp. 3-14. 21 illustrations.)

1. ROLLING OF SMALL HIGH-SPEED STEEL SECTIONS.

High speed steel presents two big difficulties from the point of view of rolling :—

(*a*) Its ductility is low and, if unevenly heated, it develops cracks and other defects.

(*b*) It is air hardening at certain temperatures.

In addition any unevenness in heating either longitudinally along the bar or across a section results in erratic filling of the groove and leads to inaccurate profiles.

The use of high frequency induction heating immediately at the entrance of the rolls gives very even heating and permits of having an exact temperature (800–850 deg. C.), at which the ductility is good, at the entrance to the rolls. Moreover, the temperature after rolling (720–680 deg. C.) is below the critical point so that no changes in metal structure are to be feared, and the rod can be re-heated for the next pass without intermediate annealing.

The Gorki works have constructed a little specialized rolling machine based on this principle in which round rods are rolled for the manufacture of drills and reamers. The rods are of 10·4 mm., 8·4, 6·6 and 4·6 mm. dia. (·41, ·33, ·26 and ·18 in.), with a tolerance of ± ·35 mm. (·014 in.) and a maximum permissible ovality of ·25 mm. (·010 in.). They have to be without defects and of a Brinell hardness of 207-255.

The machine comprises three principal items (1) variable speed gear ; (2) reduction gear ; (3) the rolls. These are all mounted in the same bedframe and operated by a 3·5 kW., 950 r.p.m. motor, giving a rolling speed of 3 r.p.m.

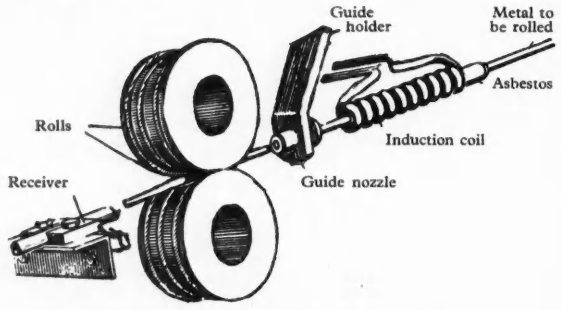


Fig. 1 **Arrangement of rolls with induction heating**

Steels of three different qualities are used. They are prepared in the form of 14 ± 1 mm. dia. (·55 in.) rods, 500 to 600 mm. (20 to 24 in.) long, forged at between 1150 deg. C. (max. at starting) and 900 deg. C. (min. at the end of operation), cooled in sand, annealed to 860 deg. C., and then sand-blasted to remove all scale and reveal possible defects.

Experiments conducted with varying temperatures of rolling showed that 730–760 deg. C. resulted in cracks and an oversize irregular section and 900–950 deg. C. again resulted in cracks, 800–850 deg. C. being ideal.

The arrangement of heating coil, guiding and receiving nozzles and the rolls is shown in Fig. 1. Their correct alignment is very important. Since quantity is relatively small, there is no automatic change-over, a batch of rods being rolled through one pass and the machine then re-set for another.

2. REPAIR OF WORN RING GAUGES BY HEAT-SHRINKAGE.

This is effected by placing the gauge inside an induction coil, as shown in Fig. 2, heating it to 700-750 deg. C. and quenching in oil. Normally a single-turn coil is used, but a long gauge may require several turns. The resulting changes of structure of the outer part of the ring lead to a reduction in volume which is sufficient to compress the otherwise unaffected inner layers, compensating for the worn-off material of the bore and permitting the ring to be used again.

The end surfaces of the ring, after shrinking, show a dark annular zone which must be concentric with the bore and be alike on both sides, otherwise the shrinkage is not even and concentric, and the gauge turns eliptical, conical or may even crack.

Fracture may also result from excessive shrinkage, so that it is essential, while heating, to watch the tempering colours closely, be guided by them in adjusting for concentricity and to drop the ring in oil when colour reaches one-third to two-thirds of the way across the section.

The coil used must have a concentrated field, otherwise excessive heating of the inner layers results, ruining their wear properties. Correct processing will leave the colour of the inner portions of the gauge unaffected.

This trouble of heating the inner layers becomes very pronounced for small gauges, below about 20 mm. (0·8 in.), which heat up in fractions of a second. These are placed within the coil and adjusted with the current off and, on switching on, are given a rapid backward and forward movement before being plunged into water for cooling down to 100–150 deg. C. and then into an oil bath where they remain for 12 hours to relieve stresses.

Female thread gauges have also been successfully treated.

The question of re-shrinking gauges which have already been treated this way, has been studied and satisfactory results obtained.

As compared with new rings this process dispenses with the making of the original blank and with the turning, substitutes thermo-shrinkage for ordinary heat treatment, cuts out grinding and leaves a similar amount of lapping. The result of applying it to various female thread gauges of 130–931 mm. (5 to 36 in.) dia. shows that a reduction of total man-hours per 100, from 800 to 130, a reduction in cost in the ratio of 7·1 to 1 and in electric power from 2000 to 195 kwh. can be achieved.

3. BRAZING OF HARD TIPS ON CUTTERS.

This requires a coil with rectangular shape of turns, made of 6 to 8 mm. ($\frac{1}{4}$–$\frac{5}{16}$ in.) outside dia. copper tube and giving clearances all round of 7 to 13 mm (·28 to ·51 in.), and not a mere 2 to 5 mm. (·08 to ·2 in.) as is sometimes stated. It has been found that with small clearances the edges of the metal sometimes melt. The length of the coil should be in excess of the length of the actual cutter by a similar amount to ensure equal

An introduction to this article deals with the properties of induction currents, choice of frequency to obtain the desired penetration, etc. Following is a description, with electrical diagrams, of two standard Russian high frequency generators used by the Gorki Works. These chapters were not considered to be of special interest and have therefore been omitted from this abstract.—EDITOR, E.D.

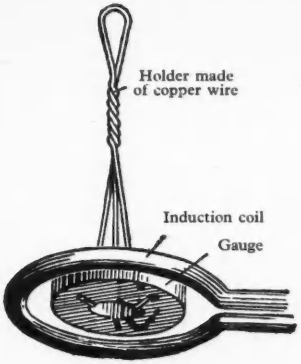


Fig. 2. **Heat shrinkage of gauges**

heating. The clearance should be a little more in height than in width to allow for the space taken up by the brazing metal and borax.

The Gorki Works have introduced this method for 600 different types of cutters, with a monthly total of about 11,000. "Pobedite"* is used, and it is found that with this technique the joint never breaks, nor does the pobedite itself crumble. Both these failures were common with gas furnace brazing.

The advantages may be summarized as (*a*) simplification (no regulation of the furnace); (*b*) speed; (*c*) precision as regards temperature; (*d*) reduced cost; (*e*) lower consumption of brazing materials; and (*f*) stronger joint owing to correct temperature and absence of oxidization.

4. SURFACE HARDENING OF TAPS.

Furnace or bath heating of taps resulted in—

(1) Distortion during cooling.
(2) Breakages in service due to excessive depth of hardening.
(3) Local losses of carbon.

Experiments were first made with 18 mm. (about $\frac{3}{4}$ in.) taps and a coil consisting of a single 25 mm. (1 in.) int. dia. turn of 14 mm. (·55 in.) tube. (With smaller diameter coil excessive surface heating resulted with insufficient heating of the core).

The tap (see Fig. 3) is threaded through the coil at a rate (depending on the hardness required) of between 2 and 12 mm./sec. (·08 and ·47 in.), while being rotated at 50 r.p.m. to ensure more even heating. The surface heats up on its passage through the coil and passes on to the cooling zone, which is about $\frac{1}{4}$ in. below the coil. Here it is sprayed by what amounts to a solid conical sheet of water which issues under pressure from $\frac{1}{16}$ in. dia. holes drilled at 45 deg. and spaced $\frac{1}{8}$ in. apart round the circumference of the induction coil. A smaller angle than 45 deg. was found to result in splash-back, giving an uneven, spotty hardened surface.

The ideal form of hardening is as shown in Fig. 4. (Rockwell Hardness Numbers given.) With similar outer hardness and different hardening conditions one can get either a very shallow hardened layer which is insufficient, or a hardened layer of excessive depth which leaves only a small round core unhardened while the outer layers show signs of overheating.

On two batches of 100 taps each, 17 mm. dia. × 110 mm. long (·67 in. × 4·3 in.) one of which was hardened by the old method and the other by high frequency induction, the following results were obtained:—

* A Russian hard alloy.

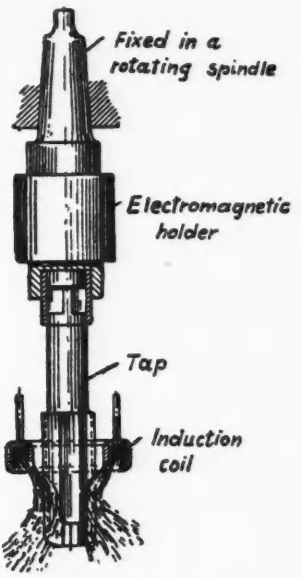


Fig. 3. **The hardening of taps**

	High Frequency.	*Furnace.*
Permissible deviation from straight	·003 in.	·003 in.
Number of taps not requiring straightening	74	15
Maximum deviation recorded ..	·005 in.	·012 in.
Maximum time for straightening one tap	1 min.	2·5 min.
Total straightening time ..	35 min.	195 min.

Having obtained excellent experimental results the Gorki Works constructed a series of single-turn coils 15, 20, 30, 35 and 42 mm. (·6 to 1·65 in.) int. dia. covering a wide range of taps with clearances varying from 2 to 7 mm. (·08 to ·28 in.).

Heating temperatures varied somewhat according to

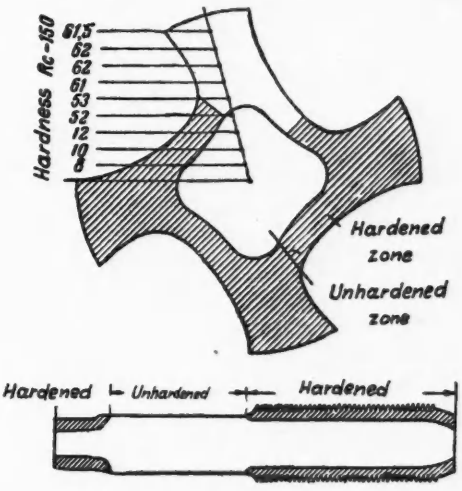


Fig. 4. **Illustrating hardening of taps**

the type of thread, the lower range of temperatures (790 to 800 deg. C.) being used for fine threads.

Subsequently multi-turn induction coils were introduced permitting simultaneous heating of the whole tap. With these, the cooling water has to be delivered from a separate spraying ring which is a yoke-like box in halves with many rows of small holes, and goes outside the induction coil, spraying both the tap and the coil when heating is completed and the water turned on.

5. SURFACE HARDENING OF HIGH SPEED TOOLS.

High speed steel has a thermal conductivity two to three times lower than carbon steel of the same carbon content. It is, therefore, widely assumed that rapid heating must inavoidably bring about internal stresses which, in view of the poor ductility of this material, are bound to cause distortion and cracks.

However, the usual salt bath method also has considerable drawbacks, and in particular the fact that it results in a reduced-carbon layer, which is very unwelcome if subsequent grinding is not desired. It was therefore decided to experiment with induction heating for milling cutters and the results have shown that while this method has its difficulties they are not due to internal stresses. On the contrary, tools of improved surface qualities can be obtained. The logic of this is as follows :—

1. The heating occurs in a thin surface layer due to the tendency of high frequency currents to stay at the surface.

2. The heating is very uniform in the layer in question : there is no tendency for it to concentrate at points of low conductivity, since each molecule is heated individually, as it were.

3. The resulting structure is uniform, with more completely dissolved carbides, which greatly reduces temperature stresses.

4. When subsequently austenite changes to martensite the internal stresses are similarly kept low.

The method used is illustrated in Fig. 5, a universal hardening machine being used to rotate the cutter inside the induction coil, and to drop it through into an oil bath when the required temperature, 1260 to 1290 deg.C. is reached. Good results were obtained with air gaps varying from 2 to 17 mm. (·08 in. to ·67 in.) and coils of a height equal to the thickness of the cutter.

The resulting structure was :—

(*a*) Hardened zone : martensite, carbides, fine needle martensite, depth of hardening 5 to 8 mm. (·2 in. to ·3 in.). Hardness : Rockwell 64 to 61.

(*b*) Transition zone : martensite and carbides. Hardness, 58 to 41 Rc.

(*c*) Remainder : unaffected, of 19 to 26 Rc. hardness.

Fig. 5. **Hardening of a milling tool**

No loss of carbon or distortion occurred.

The main difficulty of the method lies in the fact that the heating is very rapid and the desired temperature is close to the melting point. Thus it is difficult to judge temperatures precisely, and experimentally obtained timings are liable to be upset by voltage variations at the mains. Thus the cutting edge sometimes melts whilst on other occasions the hardening is found inadequate. It is thought that photo-electric methods of temperature measurements may give the necessary speed and precision. In the meantime this method of hardening must still be regarded as in an experimental stage.

FRANCE

VERTICAL FURNACE FOR THE NORMALIZING OF STRIP.

(From *Revue du Chauffage Electrique*, Vol. 14, No. 2, March/April, 1946, pp. 24-25, 2 illustrations.)

THE furnace shown in Fig. 1, for which a patent has recently been granted is under construction at a steel works in the East of France. It is designed for a temperature of 980 deg. C. and for a throughput of 300 kg per hour of mild steel strip of 0·1—1·5 mm thickness and 400 mm maximum width.

The reels are so arranged that several strips of smaller width can be passed side by side through the furnace. Furnaces of this type can be built for strip of greater width without difficulty. For local as well as economic reasons, combined electric and gas heating is provided, gas heated radiant tubes being used in the preheating zone in which the strip is heated to 700-800 deg. C. In the installation referred to the heating gas is supplied from a producer gas plant which does not allow the attainment of preheat temperatures in excess of 750 deg. C.

The strip entering at the bottom of the furnace passes first through the preheating zone in which it is heated by the radiant gas fired tubes. It is then passed through the final heating and temperature equalizing zone where it is heated by means of electrical resistor elements which raise the temperature of the strip from 700 deg. C. to some 930 deg. C. Immediately upon leaving the equalizing zone the strip enters the cooling zone in which it is abruptly cooled. This cooling zone is in the shape of a jacketed muffle with cooling water circulating through the jackets. Actually the cooling zone is served by several jackets, the individual water supply to which is so adjusted that the strip leaves the furnace at a temperature low enough to preclude its discoloration. An exit temperature of the strip of 130-150 deg. C. is chosen in order to keep the cooling zone at reasonable length. An air blower is provided to cool the sheet to about 40 deg. C. before it is wound upon the reel.

Provision is made for the use of a controlled atmosphere consisting of the products of combustion of ammonia. Drying of these combustion products is not resorted to as the presence of humidity does not affect

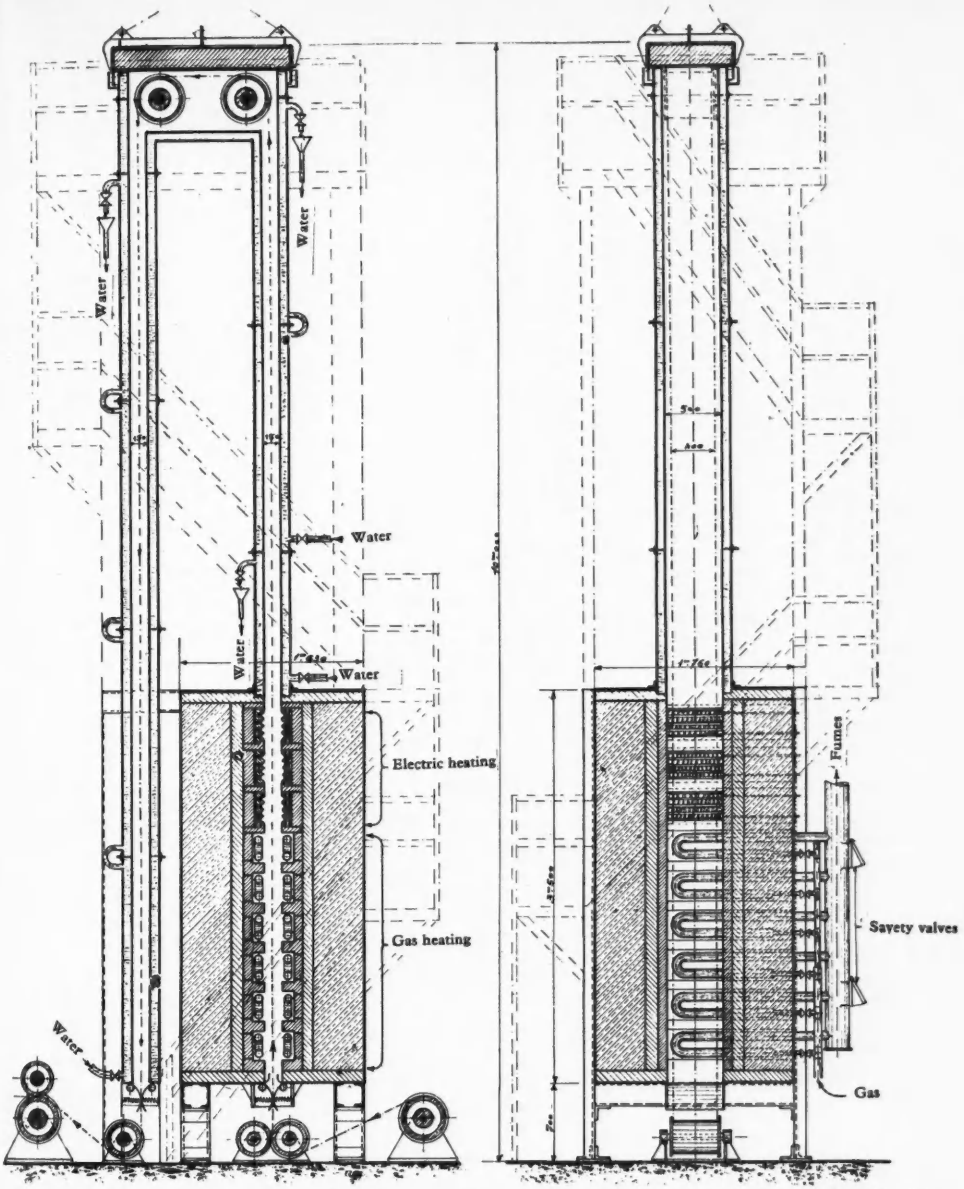


Fig. 1. **Vertical section through annealing furnace with controlled atmosphere for the normalizing of strip.**

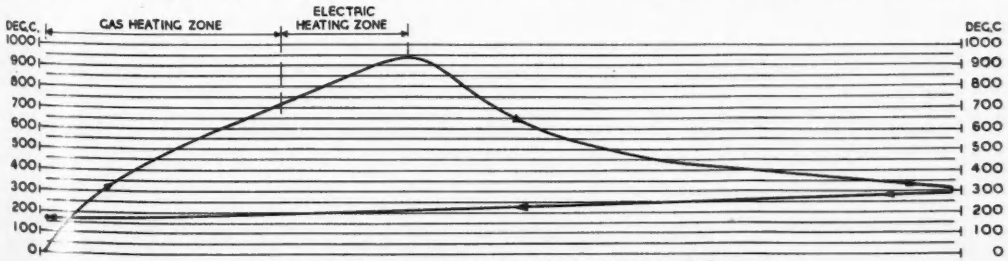


Fig. 2. **Temperature distribution in strip passing through furnace.**

mild steel strip. Special precautions are taken to prevent the escape of protective gas at the points where the strip enters and leaves the furnace.

A highly accurate automatic temperature control gear is provided, which controls solely the electrical heater part and which incorporates a pyrometer of the total radiation type measuring the temperature of the strip at a point where it leaves the equalizing zone.

As the gas-heated zone serves only to preheat the strip it is not equipped with automatic temperature control as such ; but there is a gas flow controller which is actuated by a thermocouple located in the vicinity of the radiant tubes ; and this controller acts as a protective device. The diagram given in Fig. 2 shows the temperature distribution in the strip over the distance from inlet to outlet of the furnace.

HOLLAND

THE LARGEST BUS IN HOLLAND.

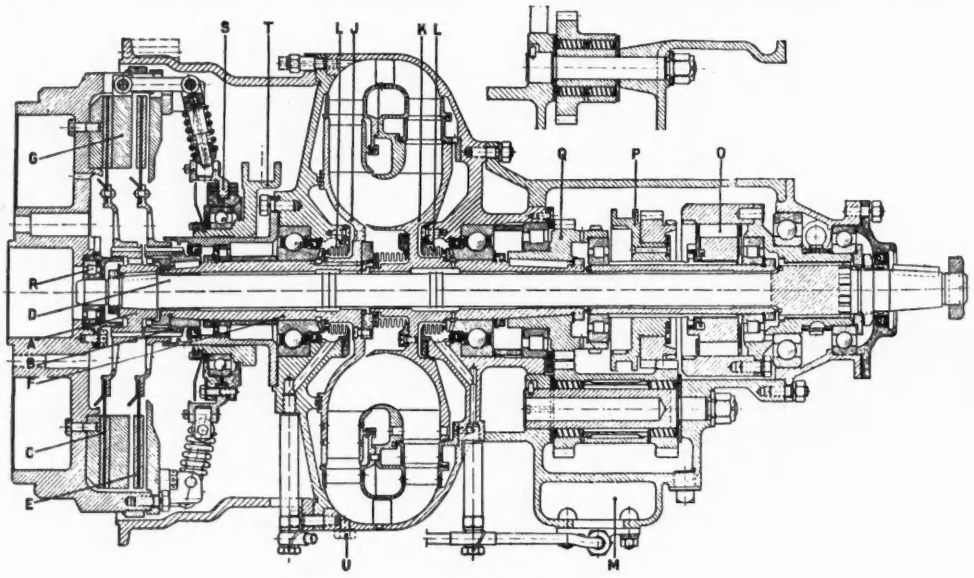
By W. B. I. HOFMAN AND P. J. DIJKSMAN. (From *De Ingenieur*, Vol. 58, No. 19, May 17, 1946, pp. W17-19, 4 illustrations.)

THE largest bus built in Holland is a vehicle of 10·58 m (34 ft. 8½ ins.) length with accommodation for 71 passengers. This vehicle was built in 1941 by the Kromhout Motor Works, Amsterdam, for the Amsterdam Municipal Transport Board. The bus was hidden away during the German occupation, and when it was finally discovered, it could not be put into service as all important parts of the engine had been removed.

The engine is a 110 hp. 6-cylinder Kromhout-Gardner diesel with a normal speed of 1850 r.p.m. The engine mounting consists in a three-point type rubber suspension. The steering mechanism is equipped with a Bosch electro-pneumatic steering gear, the compressed air being supplied by a Westinghouse compressor.

Gear changing is completely eliminated by the provision of the fluid drive shown in the accompanying illustration. This drive consists of (1) a double acting friction clutch, (2) a hydraulic system, (3) a free wheeling device interposed between the driven shaft of the hydraulic gear and the driven shaft, and (4) a reversing gear.

The two-disk type clutches C and E are mounted on the slidable hubs A and B respectively. The driving part of both clutches is represented by the clutch disk G situate between the two disks C and E. When disk C is engaged with disk G, the drive is direct without interposition of the torque converter, while the latter is put into action by engaging the disks E and G. The coupling disk C is attached to the solid shaft D, while the disk E is attached to the hollow shaft F which carries the pump wheel J of the torque converter. The pump wheel is of the usual centrifugal type. The turbine wheel is attached to a hollow shaft coupled with the driven shaft by means of the free wheeling device O. By shifting the slidable gear P to the left, the turbine wheel is coupled with the free wheeling gear ; and by shifting it to the right, the reversing gear is engaged.



A. Hub of direct-drive clutch disk.
B. Hub of converter clutch disk.
C. Clutch disk of direct drive.
D. Direct-drive shaft.
E. Clutch disk of torque converter drive.
F. Hollow shaft for converter drive.
G. Axially slidable clutch disk.
J. Pump wheel
K. Turbine wheel.
L. Seal.
M. Reservoir.
O. Free wheeling device.
P. Axially slidable gear clutch.
Q Constant mesh transmission.
R. Centre bearing.
S. Clutch muff.
T. Fork engaged by operating lever.
U. Drain.

CRITICAL VOLTAGE OF DIRECT CURRENT CORONA.

By J. STENZL. (From *Elektrotechnicky Obzor*, Vol. 34, No. 17-18, December, 1945, pp. 254-255.)

THE density of the electrical discharge current between two parallel planes can be expressed by the following equation :

$$j = j_0 \frac{(\alpha-\beta) \cdot e^{(\alpha-\beta) l}}{(\alpha-\beta) \cdot e^{(\alpha-\beta) l}} \quad .. \quad .. \quad (1a)$$

where α and β are Townsend's ionization factors, l the distance between the electrodes and j_0 the density at the cathode ($x = 0$). Putting $\alpha = k\beta$ with the constant $k \gg 1$, equation (1a) is transformed into :

$$j = j_0 \frac{(k-1) e^{\beta (k-1) l}}{k - e^{\beta (k-1) l}} \quad .. \quad .. \quad (1b)$$

The discharge becomes independent if the denominator in equation (1b) is zero, i.e. if :

$$e^{\beta (k-1) l} = k, \quad .. \quad .. \quad (2a)$$

or $$\beta l = \frac{\ln k}{k-1}, \quad \alpha l = \frac{k \ln k}{k-1} \quad .. \quad .. \quad (2b, c)$$

For any electrical field then :

$$\int_0^l \alpha dl = \frac{k \ln k}{k-1} = K, \quad \int_0^l \beta dl = \frac{\ln k}{k-1} = K' \quad .. \quad (2d)$$

The ionization factors are proportional to the pressure p of the gas, the factor of proportionality being a function of the ratio $\mathfrak{E}/p$:

$$\frac{\alpha}{p} = \varphi\left(\frac{\mathfrak{E}}{p}\right), \quad \frac{\beta}{p} = \Psi\left(\frac{\mathfrak{E}}{p}\right) \quad .. \quad (3a, b)$$

If the mean free path of an ion is λ, the average number of free paths per unit length is $1/\lambda$. A certain fraction of this total number of free paths exceeds a certain length ξ ; according to the kinetic theory this fraction equals $e^{-\frac{\xi}{\lambda}}$. If an ion travels over a distance equal to a unity of length, the average number of free paths individually exceeding in length a given ξ is :

$$N(\xi) = \frac{1}{\lambda} e^{-\frac{\xi}{\lambda}} \quad .. \quad (4)$$

Ionization of a gas particle upon collision with an electron takes place if the energy acquired by the electron from the field before its collision with the particle equals or exceeds the ionization energy V_1 of the element in question, namely :

$$\mathfrak{E}\, q_e \xi \geqq q_e V_1, \text{ or } \xi \geqq \frac{V_1}{\mathfrak{E}} \quad .. \quad .. \quad (5)$$

where q_e is the charge of the ion and $\mathfrak{E}$ the field strength. Assuming that all collisions complying with condition (5) bring about the corresponding ionization, then :

$$N(\xi) = \alpha = \frac{1}{\lambda} e^{-\frac{V_1}{\mathfrak{E}\lambda}} \quad .. \quad (6)$$

Now as $\lambda p = \text{constant} = \frac{1}{A'}$; $\frac{1}{\lambda} = A'p$ and therefore :

$$\frac{\alpha}{p} = A' e^{-\frac{B'}{\mathfrak{E}/p}}, \quad .. \quad .. \quad (7)$$

where $V_1 A' = B'$. If the pressure p is substituted by the density δ of the ionized gas which is proportional to it (for air $\delta = 1$ at $p = 76$ cm Hg, and $t = 25$ deg. C), equation (7) is transformed into :

$$\alpha = A \delta e^{-\frac{B\delta}{\mathfrak{E}}}, \quad .. \quad .. \quad (8)$$

where A and B are constants for the particular gas.

Now equation (8) can be applied to the case of coaxial cylindrical electrodes. For these the following equation is valid :

$$\mathfrak{E}_r = \frac{1}{r} \mathfrak{E}_0 r_0 \quad .. \quad .. \quad (9)$$

where r_0 is the radius of the inner electrode.

Introducing equations (8) and (9) into (2d) :

$$K = \int_{r_0}^{R} \alpha_{(r)}\, dr = A\delta \int_{r_0}^{R} e^{-\frac{B\delta}{\mathfrak{E}_0 r_0} r} \cdot dr =$$

$$= \frac{A}{B} \mathfrak{E}_0 r_0 \left[e^{-\frac{B\delta}{\mathfrak{E}_0}} - e^{-\frac{B\delta}{\mathfrak{E}_0}\frac{R}{r_0}} \right] \quad (10)$$

where R is the radius of the outer electrode.

With a sufficiently high gas density and with $R \gg r_0$, the second half of the equation can be neglected :

$$K = \frac{A}{B} \mathfrak{E}_0 r_0 e^{-\frac{B\delta}{\mathfrak{E}_0}} \quad .. \quad (11)$$

wherefrom :

$$r_0 = \frac{BK}{A} \cdot \frac{1}{e^{-\frac{B\delta}{\mathfrak{E}_0}} \cdot \mathfrak{E}_0} = \frac{BK}{A} \cdot \frac{1}{B\delta} \cdot \frac{1}{e^{-\frac{B\delta}{\mathfrak{E}_0}} \cdot \frac{\mathfrak{E}_0}{B\delta}} \quad (12)$$

Introducing $C = \frac{BK}{A}$, $x = \frac{\mathfrak{E}_0}{B\delta}$

$$r_0 = \frac{C}{B\delta} \frac{1}{e^{-\frac{1}{x}} \cdot x} \quad .. \quad .. \quad (13)$$

To simplify equation (13) it can be assumed that, for practical purposes, only the following conditions are of importance :

$x \leqq 0{\cdot}35$, $B \leqq 230$, $\delta = 1$, $\mathfrak{E}_0 \leqq 80{\cdot}5$ kV.

in this case the curve representing the expression $y = e^{-\frac{1}{x}} \cdot x$ can with sufficient accuracy be substituted by the parabola $y = \frac{(x-a)^2}{2p}$ where $a = 0{\cdot}15$ and $2p = 2$.

Equation (13) becomes thereby :

$$r_0 = \frac{C}{B\delta} \cdot \frac{2p}{(x-a)^2},$$

hence : $$x = a + \sqrt{\frac{2pC}{B}} \cdot \sqrt{\frac{1}{\delta r_0}}$$

and further $\mathfrak{E}_0 = aB\delta + \sqrt{2pCB}\,.\sqrt{\frac{\delta}{r_0}}$.

The constants B and C can be determined experimentally for various gases. For air, the corresponding figures are : $B+ \simeq 184$, $B- \simeq 230$, $C+ \simeq 0{\cdot}254$, $C- \simeq 0{\cdot}241$, and the critical potential for the positive wire :

$$\mathfrak{E}_{0+} = 27{\cdot}6\delta + 9{\cdot}7\sqrt{\frac{\delta}{r_0}}\ [\text{kV/cm, cm}] \quad .. \quad (14a)$$

and for the negative wire :

$$\mathfrak{E}_{0-} = 34{\cdot}5\delta + 10{\cdot}3\sqrt{\frac{\delta}{r_0}}\ [\text{kV/cm, cm}] \quad .. \quad (14b)$$

which is in good agreement with the empirical formulae developed by Peek ; these have been modified and are quoted differently by various authors, for instance :

$$\mathfrak{E}_{0-} = 35{\cdot}0\delta + 8{\cdot}06\sqrt{\frac{\delta}{r_0}} \text{ and } \mathfrak{E}_{0-} = 33{\cdot}9\delta + 10\sqrt{\frac{\delta}{r_0}}. \quad (15a,b)$$

It must, however, be borne in mind that a number of simplifications have been introduced in the preceding deductions. To arrive at equation (8) it has been assumed that after collision with a gas molecule the electron resumes its flight with an initial energy equal to zero and furthermore that each collision complying with the conditions of equation (5) produces an ion. Neither of these assumptions is absolutely correct. The assumption $\alpha = k\beta$ is valid only for discharges where no polarizing effects occur, i.e. where the critical field strength does not depend on which of the electrodes is positive and which negative. At normal atmospheric pressure this condition is fulfilled to a sufficient degree except where thin wires and sharp edges are concerned. K in equation (2d) is not absolutely constant, but is smaller for larger values of $\mathfrak{E}/p$ and vice versa.

If, notwithstanding the many simplifications, such good agreement is reached between the theoretical and empirical equations this may be attributed to individual errors cancelling each other ; however, it may still be taken as a proof of the validity, in principle, of Townsend's theory of ionization by collision.

SWEDEN

HIGH FREQUENCY FURNACE FOR THE PRODUCTION OF HIGH STRENGTH CAST IRON.

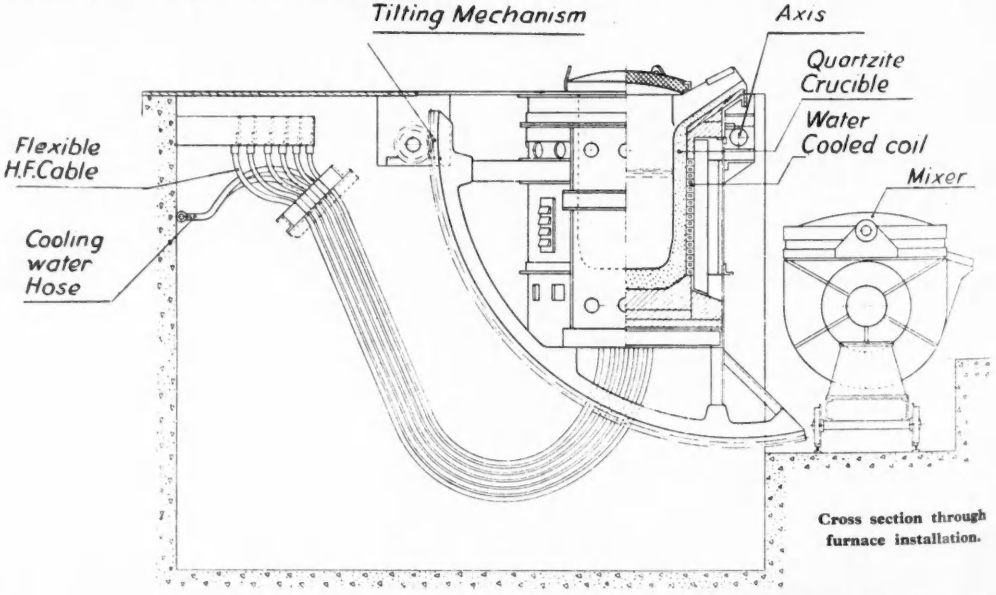
By I. Forslund. (From *Gjuteriet*, Vol. 36, No. 5, May, 1946, pp. 74-81, and 92, 10 illustrations.)

Because of the fuel shortage experienced in Sweden during the War the SKF Works decided upon the installation of a high frequency melting furnace for the production of high-strength cast iron. This furnace, which came into operation in July, 1943, is outlined in the accompanying illustration. Normal furnace capacity is 5·5 tons, but charges of 6 tons can be dealt with, although with a higher power consumption per ton of charge. The tilting mechanism is shown in the illustration.

The furnace coil consists of water cooled copper tubing insulated by means of a high heat resistant ceramic material. Magnetic screening of the coil is effected by a laminated sheet metal casing, which also serves to support the coil. The cooling water consumption amounts to 220 litres per minute. In order to preclude the possibility of condensation of atmospheric moisture upon the cooling coil, the admission temperature of the cooling water is kept constant at 35 deg. C. A portion of the cooling water which is discharged at 50 deg. C., is mixed with the cold supply water to maintain the aforementioned inlet temperature of 35 deg. C.

The motor-generator set consists of a three-phase induction motor of 1850 h.p. coupled with a generator delivering 1250 KW at 2500 Volt and a frequency of 743 cycles. An automatically controlled static condenser maintains the power factor as near unity as possible. Both the condenser and the motor generator set are air cooled.

Furnace output averages two tons of molten metal per hour, about 33 tons being produced in two shifts per day.



Cross section through furnace installation.

JET PROPULSION.

By W. F. DURAND. (From *Mechanical Engineering*, Vol. 68, No. 3, March 1946, pp. 191-193).

IT will be understood that the device considered in the present discussion comprises three essential elements :

1 An air compressor.
2 A combustion chamber.
3 A gas turbine.

To these items may be added an exhaust passage leading from the turbine to the outside air, and provided with some device for varying the size of the exhaust opening in accordance with the conditions of operation of the device as a whole.

PRINCIPLES OF JET OPERATION.

The basic fact which determines the production of a propulsive thrust is the production of a sternward acceleration in a certain mass, the reaction from which gives the forward thrust desired. Defining the useful work as the product of this thrust into the forward motion of the object, the ratio of this useful work to the total energy of the jet is termed the propulsive efficiency.

Suitable analysis of the mechanics of the problem shows that the propulsive efficiency is directly dependent upon the ratio of the speed of discharge of the products of combustion to the speed of advance of the object (aeroplane). Normally the former of these two speeds is much larger than the latter. If, however, they could be made equal, the efficiency would be unity or 100 per cent. However, the same analysis shows that the increment of velocity impressed on the mass of gas discharged is the difference between these two velocities, which we may designate by u and V, the former the speed of discharge of the gas, relative to the plane, the latter the speed of the plane relative to the earth. The increment of velocity impressed on the discharged gas is then $(u—V)$. If, then, u and V are equal, the increment is zero and the propulsive thrust vanishes. We thus have the ideal limit condition of an efficiency of 100 per cent. with the vanishing of the forward thrust. Actually, the propulsive thrust is measured by the product of the mass of gas discharged per second multiplied by the increment in velocity or in symbols by $M\ (u—V)$.

But the resistance of the plane in flight must be exactly balanced by the propulsive thrust, which as just noted is measured by the product $M\ (u—V)$. Hence, at any given speed of advance, the value of M must vary inversely as $(u—V)$, that is, the smaller $(u—V)$ and the higher the efficiency, the greater must be the mass of gas discharged to produce the needed thrust. And so, at the limit of u equals V, $(u—V)$ becomes zero, and M would need to be infinitely great to give the needed propulsive thrust. Put into plain language, this means that for any given value of the thrust, the mass of gas discharged must be greater and greater as V approaches u in value, with the corresponding increase in propulsive efficiency. But increase in the mass of gas discharged, of which by far the major part is air, means increase in the size and weight of the equipment, as well as lower temperature of the gases acting on the gas turbine.

Lying ahead of the propulsive efficiency is the thermodynamic efficiency, covering the transformation of heat into jet energy and mechanical work. Here, the principle of Carnot enters with the condition that the thermodynamic efficiency will increase the higher the temperature limit of the process. This means that higher thermodynamic efficiency calls for higher temperature of the gas acting on the gas turbine.

This complex situation calls, obviously, for a compromise between size and weight, thermodynamic efficiency and propulsive efficiency.

On the other hand, as the speed of the plane decreases relative to that of the gas discharge, the factor $(u — V)$ increases and the thrust per pound of gas increases, while the propulsive efficiency decreases.

However, the value of the gas discharge velocity u is naturally high. The mechanics of the situation tend toward velocities well beyond normal aeroplane speeds. It results that with an aeroplane speed of normal commercial value, for example, 300 to 400 ft/sec., with gas discharge of 1200 to 1500, the propulsive efficiency will be low.

The practical result is that jet propulsion suffers in propulsive efficiency at low plane speeds, and only reaches acceptable values at the highest speeds obtainable under present conditions. These present highest speeds, moreover, are more easily realized at high altitudes than at low, so it results that the normal field for jet propulsion, the field in which it most nearly realizes its potential superior performance, is at high speeds and high altitudes.

RESEARCH PROJECTS IN JET PROPULSION

Turning now to another phase of the broad subject of jet propulsion, we may inquire as to the present outstanding problems. The directions in which research and study should be directed in order to bring about improvement in this new form of propulsive engine may be denoted :

1 Problems of combustion.
2 Problems of materials of construction.
3 Problems of design.
4 Problems of operation.
5 Problems of supersonic speed.

Combustion. The problem of combustion comprises the entire field of the combustion of a fluid fuel under the conditions prevailing in a jet-propulsion engine, that is, in a rapidly moving current of air in quantity greatly in excess of the amount required to furnish the oxygen needed for combustion. The ends sought are as follows :

1. The maintenance of active combustion under all rates of fuel flow as determined by the throttle opening.
2. The complete combustion of the fuel within the minimum volume and within the limits of the combustion chamber itself.
3. Assured ignition and pickup of combustion from shutdown of the motor or from idling at any altitude and any speed of the plane.

Before oxygen can unite with hydrogen or carbon, the latter two substances must be in the free molecular or atomic state. This means that the fuel must pass from the liquid to the vapour phase as a preliminary to the phenomena of combustion. There are therefore available two techniques for the introduction of the fuel into the combustion chamber :

(*a*) Introduction in the liquid phase in the form of a fine spray, the particles of which flash promptly into vapour, and the molecules in vapour form proceed to break up into their constituent atoms which are then free to unite with the oxygen.

(*b*) Vaporization of the fuel outside of the combustion chamber and introduction of the fuel in the vapour phase into the chamber and into contact with the oxygen carried by the air.

Both of these techniques have been tried, the former appearing on the whole to have, so far, given the most satisfactory results.

There remains, however, a wide field of research directed toward improvement in both of these methods of fuel introduction into the zone of combustion.

The problem of flame propagation is also a phase of the problem of combustion needing further study. If the zone of combustion is to be limited to desirable lengths, the velocity with which the flame can spread through a mass of air and fuel vapour is a determining factor of the volume within which, with the rapidly moving current of air, the combustion can be completed.

The maintenance of combustion in the rapidly moving current of air is a condition which has presented trouble to meet. Some form of baffle or shielding of the actual

inlet point of the fuel entrance has been found needful. The best form and arrangement of such shield or baffle is a problem well worthy of further study and trial.

Materials of Construction.—The major problems under this head relate to a search for high-temperature-resisting metals with other properties suited to the making of turbine blades. Reference has been made to the thermodynamic efficiency as dependent on the upper gas temperature; the higher the temperature the higher the efficiency.

In 1941, when special attention became directed to the matter of high-temperature-resisting metals, 1200 or 1300 deg. F, was about the limit which could be attempted with the best materials that metallurgical science could provide. During the years of the war, active research was carried on, with the result that the upper limit of gas temperature was raised some 400 to 500 deg. F. This, however, should be considered only as a way station, and active search should continue for new combinations of metals carrying the temperatures to still higher levels.

There are at least two other directions in which search should be made for dispositions permitting the use of higher gas temperatures. These are the use of ceramic-coated blades and the cooling of the blades.

Ceramic materials in themselves are lacking in the physical properties of strength for use as the sole material for the blades of a gas turbine. But the combination of a ceramic coating to take the impact of the hot gas backed by a metal for the needed strength, appears to offer definite hope. The principal problem in such a construction is the finding of two materials, metal on the one hand and a ceramic on the other, with a sufficiently high melting point, which will bond with the metal and with a coefficient of expansion sufficiently near that of the metal to insure against cracking and flaking off under the wide range of temperatures to which the blades are subjected.

For cooling of the turbine blades, three methods have received some study. In one method, the blades are made hollow and adapted to the flow of cooling air outward through them. In a second method, the blades, in the course of a revolution, pass through a sector of the circle in which is cooling air instead of hot gas. This method was employed in certain gas turbines of German design with what were claimed as very satisfactory results.

Still another method of cooling has been proposed, which provides for the flow of a cooling liquid (water or liquid fuel) through a passage in the root of the turbine blade. This method depends on the flow of heat from the blade or vane itself to the relatively more massive root, encased in the turbine disk and carrying a passage for the flow of the cooling fluid. The idea appears to present distinct promise.

Design.—Design is a broad term covering materials, form, and arrangement of parts as well as individual function. Jet-propulsion engines represent a combination of many materials, diverse forms, with multiple arrangement possibilities and functions. It is by no means sure that the present conventional forms of such engines are the best possible. Rather, it is to be expected that new materials, other forms, and arrangements of parts may well present marked improvement over those characteristic of the present-day types and forms.

Improvement in turbine-blade materials has already been singled out for special mention. In addition there is the problem of attaching the blade to the turbine disk, the problem of the main bearings under the high rotative speeds (10,000 to 20,000 r.p.m.) characteristic of jet-propulsion equipment; the problem of the best material for the shell of the combustion chamber, as well as the problem of the best size and arrangement of openings in the shell of the chamber for the admission of the air; the problem of the best mode for handling the excess air in such way as to provide needed cooling effects on parts of the equipment in contact with the hot gas; the problem of the best form and arrangement of variable partial closure for the exhaust tail pipe and of the control of this valve by the pilot; the aerodynamic problems of the best forms of the rim of the entrance opening at the nose of the engine, of the form and arrangement of the burners within the combustion chamber in such fashion as to present the minimum loss of pressure in the air flow through the chamber, and the best form and proportions of the tail pipe.

Operation.—The proof of a jet-propulsion engine is in its performance in the air. A perfect performance implies perfect and prompt following of the throttle up and down, continous and regular operation under all conditions of speed and altitude with immediate pickup from engine-off condition under any altitude, attitude and speed of the plane.

Jet-propulsion engines at the present time hardly meet this specification of performance. There may be irregularity in the combustion in following the throttle over a certain part of its range of operation. There are zones of operation, as defined by speed and altitude, within which it is difficult or impossible to start the engine from the engine-off condition. This opens a broad field of development research and study.

Supersonic Speeds.—The chief condition which gives significance to jet propulsion as a substitute for the conventional propeller is the freedom of the former from loss of efficiency under speeds closely approaching, or equal to, or beyond the speed of sound in air—about 1140 ft/secs. at sea level. The conditions are those of gradual approach and begin to appear in evidence at speeds around 70 per cent of the speed of sound.

However, while jet propulsion in itself is free from loss of effect or efficiency at speeds subsonic, sonic, or supersonic, the same is far from true for the remainder of the aeroplane. This means that at speeds at which jet propulsion can be expected to come into its own as a means of aeroplane propulsion, the aeroplane as a whole and in all its parts is subject to the special conditions arising at these high speeds.

SONIC AND SUPERSONIC SPEED STUDIES NECESSARY.

The aerodynamics of low and present commercial speeds is well understood in theory and has been the subject of thorough-going research in all the leading countries of the world. Only more recently has the importance of the aerodynamics of sonic and supersonic speeds forced itself on the attention of designers and aeronautical engineers.

At sonic and supersonic speeds there is formed in the air in contact with the moving object a shock or compression wave which travels with the object and modifies in extreme degree what may be called its aerodynamic form. The laws of fluid flow around the object are profoundly modified, the resistance or drag of the object is greatly increased, and the distribution of pressure over its surface undergoes extreme changes. Due to this latter condition, control surfaces properly adapted for use at normal speeds may fail utterly to furnish control at sonic and supersonic speeds.

All of this means that hand in hand with the study of ways and means for the improvement of the jet-propulsion engine in itself, there must go continuing and active research and study of the aerodynamics of speeds through the subsonic, through the sonic, and on into the supersonic.

Owing to the large amounts of power required for the operation of wind tunnels at these high speeds, much or most of the work thus far done by way of direct laboratory experimentation has been on small-scale models and under the conditions of two-dimensional flow. This does not tell the whole story and what is needed is larger wind tunnels with larger models tested under the conditions of three-dimensional flow.

ELIMINATION OF IMPACT LOADS ON THE TRUNNIONS OF AN ELECTRO-MAGNETIC CONTACTOR.

By I. S. SOLODUHO. (From *Vestnik Electro-Promishlennosti*, No. 3, 1946, pp. 10-13, 7 illustrations.)

A CONTACTOR is one of the commonest component parts of an automatic circuit and it usually has heavy duty to perform : frequent making and breaking, sometimes up to 1500 times per hour, 24-hour-a-day service, and currents which may instantaneously be many times in excess of their nominal figure.

The fastest wearing parts, as measured by length of service, are the contacts themselves. But the length of service of the mechanism as a whole is determined by wear of other parts which are less easy to replace, and amongst these trunnions and trunnion bushes are very prominent and showed appreciable wear after relatively few operations when several types of contactors were tested at the Kharkov Electrical Institute. When the number of operations reached the order of 1 to 2 millions the wear at these points was so great that the contactors could scarcely be described as in working order.

Amount of friction and frictional wear, of course, are dependent on a great number of factors like type of friction (rolling, sliding, etc.), lubrication, relative velocity and acceleration of parts, cleanliness and nature of surface. But in contactors the most prominent features causing wear are :—

(1) A rotary motion starting from zero rising to a maximum and falling to zero again, and then reversing. This is a particularly unfavourable form of movement because it involves a lot of overcoming of static friction which is greater than kinetic friction.

(2) Impact loading.—From the above it follows that lubrication cannot be fully effective, because there is no hydro-dynamic (" Michelling ") effect at starting and stopping, i.e., precisely when the forces are biggest and in addition the impact blows tend to drive out the lubricant and make surfaces almost dry. Wear in service is greatly increased by the fact that abrasive matter finds its way between the surfaces.

It is therefore the aim of the present study to consider the impact forces in their relation to wear, to suggest a method of calculating them and to indicate how they may be eliminated.

THE ORIGIN OF IMPACT FORCES.

The typical working cycle of a contactor is as follows :—

(1) Motion of the whole moving system, up to the moment of contact. The touching of the contacts results in a dynamic force and if there is vibration in the system this is followed by a further series of such forces, gradually dying out. A typical contactor, at the moment of contact, is shown in Fig. 1.

(2) Further movement of the system, with contacts rolling against each other until the main lever strikes the core of the electro-magnet. Here again repeated blows are possible if there is vibration.

(3) When " breaking ", at the moment when contacts cease to roll there is a blow of the contact system against the main lever along the line *R*.

(4) Remainder of the " breaking " movement, ending in the main body striking the stationary stop.

As a result of the impact between contacts there is a reaction S_2 at axis O_2 which is transmitted along the lever and results in a reaction at O_1. Similar reactions occur at the end of phase 3. Phases 2 and 4 result in reaction at O_1 only. Thus O_1 experiences four successive blows during each cycle.

The method described below permits of calculating these reactions, the example taken being the worst offender, phase 1.

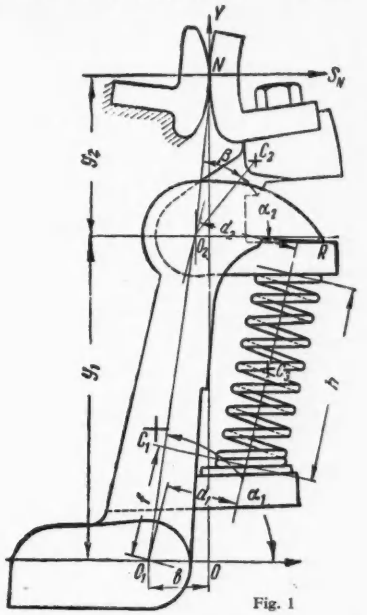


Fig. 1

CALCULATION OF IMPACT REACTIONS.

Consider the system shown in Fig. 2, with the main lever 1, articulated contact bracket 2 and spring 3 ; the weight of the latter can be neglected as a first approximation.

The system can very well be regarded as in one plane and if co-ordinates are chosen so that axis OY is tangential to the point of first contact the calculation is simplified.

If m is mass ; u_{ox} and u_{oy} components parallel to the two axes of the linear velocity of the centre of gravity before the blow ; u_x and u_y velocity components after the blow, J_c moment of inertia about the centre of gravity, ω_0 original angular velocity and ω angular velocity after blow, then :

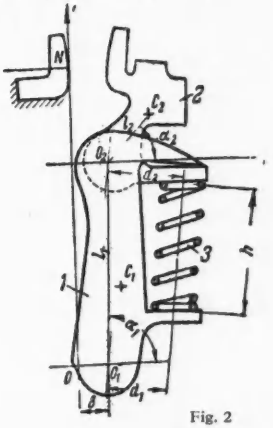


Fig. 2

$$m(u_x - u_{ox}) = \sum_{i=1}^{n} S_{1x} \quad .. \quad (1a)$$

$$m(u_y - u_{oy}) = \sum_{i-1}^{n} S_{1y} \quad .. \quad (1b)$$

$$\mathcal{J}_c(\omega - \omega_0) = \sum_{i=1}^{n} M(S_1) \quad .. \quad (2)$$

where $\sum_{i-1}^{n} S_{1x}$ and $\sum_{i-1}^{n} S_{1y}$ are the sums of external forces resolved in the direction of the two axes and $\sum_{i=1}^{n} M(S_1)$ is the sum of external moments about the centre of gravity.

In the present case (see Fig. 3), the external forces will be S_N, S_1 and S_2. And S_N will have no Y component owing to our choice of co-ordinates.

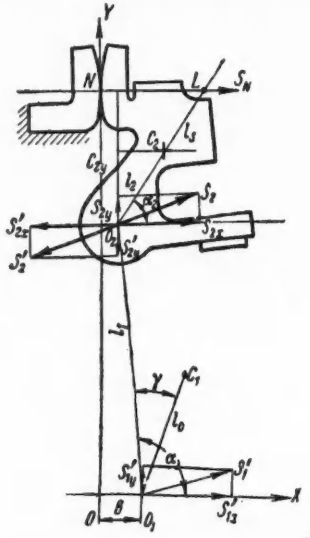


Fig. 3

Position of main lever at any moment is fully determined by the angle α_1 while that of the contact bracket is determined by α_1 and α_2.

Co-ordinates of C_2, (C. G. of the contact bracket) are :—

$$\left.\begin{aligned} x_{c2} &= b + l_1 \cos\alpha_1 + l_2 \cos\alpha_2 \\ y_{c3} &= l_1 \sin\alpha_1 + l_2 \sin\alpha_2 \end{aligned}\right\} \quad .. \quad (3)$$

The components of the velocity of C_2 can be obtained by differentiating these with respect to time and from (1a) and (1b) we obtain :

$$m_2[-l_2 \sin\alpha_2(\omega_2 - \omega_{10}) - l_1 \sin\alpha_1(\omega_1 - \omega_{10})] = S_N + S_{2x} \quad .. \quad .. \quad (4)$$

$$m_2[l_2 \cos\alpha_2(\omega_2 - \omega_{10}) + l_1 \cos\alpha_1(\omega_1 - \omega_{10})] = S_{2y} \quad (5)$$

$$\mathcal{J}_{c2}(\omega_2 - \omega_{10}) = -S_N l_8 \sin\alpha_2 + S_{2x} l_2 \sin\alpha_2 - S_{2y} l \cos\alpha_2 \quad .. \quad .. \quad (6)$$

Where ω_{10} is angular velocity of the whole system before contact, ω_1 — that of lever after contact ω_2 — that of bracket after contact.

Similarly for the lever, the equations are :

$$m_1 l_0 \sin(\alpha_1 - \gamma)(\omega_1 - \omega_{10}) = S'_{2x} - S'_{2x} = S_{2x} - S_{1x} \quad (7)$$

$$m_1 l_0 \cos(\alpha_1 - \gamma)(\omega_1 - \omega_{10}) = -S'_{2y} + S'_{1y} = -S_{2y} + S_{1y} \quad .. \quad .. \quad (8)$$

$$\mathcal{J}_1(\omega_1 - \omega_{10}) = S_{2x} l_1 \sin\alpha_1 - S_{2y} l_1 \cos\alpha_1 \quad .. \quad (9)$$

Clearly $S'_{2x} = S_{2x}$ and $S'_{2y} = S_{2y}$ but even so we have six equations with seven unknowns :

S_N, S_{2x}, S_{2y}, S'_{1x}, S'_{1y}, $(\omega_1 - \omega_{10})$ and $(\omega_2 - \omega_{10})$.

This is remedied by bringing in the fact that the ratio of the velocities of point N before and after impact is equal to the coefficient of restitution, ϵ.

$$\frac{u_{Nx}}{u_{Nox}} = \frac{y_1\omega_1 + y_2\omega_2}{(y_1 - y_2)\omega_{10}} = -\epsilon$$

where

$y_1 = l_1 \sin\alpha_1$ and $y_2 = l_N \sin(\alpha_2 + \beta) = (l_2 + l_8)\sin\alpha_2$.

From this we can get :

$$y_1(\omega_1 - \omega_{10}) + y_2(\omega_2 - \omega_{10}) = -(1 + \epsilon)(y_1 + y_2)\omega_1^0 \quad (10)$$

Solving these equations and putting :

$$\left.\begin{aligned} a_{11} &= \mathcal{J}_1 + m_2 l_1^2 \\ a_{12} &= m_2 l_1 l_2 \cos(\alpha_1 - \alpha_2) \\ a_{22} &= \mathcal{J}_2 = \mathcal{J}_{c2} + m_2 l_2^2 \end{aligned}\right\} \quad .. \quad (11)$$

We get :

$$\omega_1 - \omega_{10} = -\frac{(a_{22}y_1 - a_{12}y_2)(y_1 + y_2)}{y_1(a_{12}y_2 - a_{22}y_1) + y_2(a_{12}y_1 - a_{11}y_2)}(1+\epsilon)\omega_{10} \quad (12)$$

$$\omega_2 - \omega_{10} = \frac{(a_{11}y_2 - a_{12}y_1)(y_1 + y_2)}{y_1(a_{12}y_2 - a_{22}y_1) + y_2(a_{12}y_1 - a_{11}y_2)}(1+\epsilon)\omega_{10} \quad (13)$$

$$S_N = \frac{(a_{11}a_{22} - a_{12}^2)(y_1 + y_2)}{y_1(a_{22}y_1 - a_{12}y_2) + y_2(a_{11}y_2 - a_{12}y_1)}(1+\epsilon)\omega_{10} \quad (14)$$

$$S_{2x} = m_2[-l_2 \sin\alpha_2(\omega_2 - \omega_{10}) - l_1 \sin\alpha_1(\omega_1 - \omega_{10})] - S_N \quad (15)$$

$$S_{2y} = m_2[l_2 \cos\alpha_2(\omega_2 - \omega_{10}) + l_1 \cos\alpha_1(\omega_1 - \omega_{10})] \quad (16)$$

$$S'_{1x} = S_{2x} - m_1 l_0 \sin(\alpha_1 - \gamma)(\omega_1 - \omega_{10}) \quad .. \quad .. \quad (17)$$

$$S'_{1y} = S_{2y} + m_1 l_0 \cos(\alpha_1 - \gamma)(\omega_1 - \omega_{10}) \quad .. \quad .. \quad (18)$$

When these equations were applied to actual contactors under test, it was found that the reactions are very appreciable and their calculated direction checked up very well with the direction of greatest wear of the trunnions.

CONDITIONS UNDER WHICH IMPACT FORCES ARE ELIMINATED.

Assume absence of impact reaction at O_2 i.e. $S_{2x} = S_{2y} = 0$. Then from (9) we get $\omega_1 - \omega_{10} = 0$ and from (17) and (18) $S'_{1x} = S'_{1y} = 0$. Thus if there is no impact load at O_2 there is likewise none at O, and the main lever does not change its angular velocity.

From (4) and (5) we now get :

$$S_N + m_2 l_2 \sin\alpha_2(\omega_2 - \omega_{10}) = 0 \quad .. \quad .. \quad .. \quad (19)$$

$$m_2 l_2 \cos\alpha_2(\omega_2 - \omega_{10}) = 0 \quad .. \quad .. \quad .. \quad (20)$$

and (20) can only be true if :

(a) $\omega_2 - \omega_{10} = 0$ which can be proved to be contrary to the physical requirements of the apparatus when $(\omega_1 - \omega_{10})$ is also equal to 0.

or (b) $l_2 = 0$, in which case (19) gives $S_N = 0$, which again is contrary to physical necessities.

or (c) $\cos\alpha_2 = 0$ i.e. $\alpha_2 = 90$ deg. and $\sin\alpha_2 = 1$.

The last is the true essential condition of working without impact load on the trunnion and it is quite possible to achieve it in practice. It means, from (11) that $a_{12} = m_2 l_1 \sin\alpha_1$ and that $y_2 = l_N \cos\beta = l_2 + l_8$.

Also, from (12) and (14) we can get :

$$\omega_1 - \omega_{10} = \frac{S_N (a_{12} y_2 - a_{22} y_1)}{a_{11} a_{22} - a_{12}^2}$$

and hence $a_{12} y_2 = a_{22} y_1$. (11) now gives

$m_2 l_2 (l_2 + l_s) = \mathcal{J}_{c_2} + m_2 l_2^2$ and $\mathcal{J}_{c_2} = m_2 l_2 l_s$ .. (21)

If now we put ρ_{c_2} for the radius of gyration of the contact bracket then $\mathcal{J}_c = m_2 \rho^2_{c_2}$ and from (21)

$$\rho_{c_2}^2 = l_2 l_s \quad .. \quad .. \quad (22)$$

Given l_2 m_2 and $\mathcal{J}_{c_2}$ then we can find $l_s = \frac{\mathcal{J}_{c_2}}{m_2 l_2}$ and

$y_2 = l_2 + \frac{\mathcal{J}_{c_2}}{m_2 l_2} = l_2 \left(\frac{1 - \mathcal{J}_{c_2}}{m^2 l_2^2} \right)$ which, combined with an α_2 of 90 deg. completely defines the required position of contact bracket and of point N at the moment of contact.

Equation (19) permits of calculating S_N under these conditions.

It is interesting to see to what extent existing contactors approach the ideal requirements. Two Russian standard types were checked with the following results :

Type	$\sin \alpha_2$	$\frac{\rho_{c_2}^2}{l_2 l_s}$	$\frac{S_2}{S_N}$
MK—601—G2	·773	·765	0·6
MP—4	·972	·705	0·26

Similar calculations apply to other phases of the movement of the contactor and " no impact " conditions can be found.

ITALY

COMPUTATION OF LENGTH AND TIME OF LANDING RUN WITH AIRSCREW-BRAKING.

By S. R. GRAZI. (From *Rivista Aeronautica*, Vol. 22, No. 5/6, May/June, 1946, pp. 277-281, 6 illustrations.)

THE braking thrust of an airscrew is a function of the speed of the airscrew, of the blade dimensions and pitch, and of the relative air speed. The forces acting upon the plane during the landing run are :

Braking thrust of the airscrew : T
Aerodynamic resistance : $R_a = C_r \rho S v^2$
Rolling resistance : $R_s = C_p \rho S v^2$
The total retarding force is therefore :

$$R = T + R_a + R_s$$

The deceleration d to which the aircraft is subject is given by :

$$d = \frac{Rg}{Q} = \frac{T + R_a + R_s}{m}$$

where $m = \frac{Q}{g}$ is the mass of the aircraft.

As it is impossible to arrive at an analytical integration of the resultant differential equations, the graphical method must be resorted to. For this purpose we recall that $dV/dt = d$ and $dV/ds = d/V$. We obtain by substitution :

$$m \frac{dV}{dt} = T + R_a + R_s$$

and

$$m \frac{VdV}{ds} = T + R_a + R_s$$

and by combining with equations (1) and (2) :

$$m \frac{dV}{dt} = T + C_r . \rho . S . V^2 + f(Q - C_p . \rho . S . V^2) =$$

$$= T + f . Q + (C_r - f . C_p) \rho . S . V^2$$

By dividing by $Q = C_p . \rho . S . V_a^2$, where V_a is the landing speed, it becomes :

$$\frac{dV}{dt} = g \left[\frac{T}{Q} + f \left(\frac{C_r}{C_p} - f \right) \frac{V^2}{V_a^2} \right]$$

and

$$\frac{VdV}{ds} = g \left[\frac{T}{Q} + f + \left(\frac{C_r}{C_p} - f \right) \frac{V^2}{V_a^2} \right]$$

Solving for dt and ds we obtain :

$$dt = \frac{dV}{g \left[\frac{T}{Q} + f + \left(\frac{C_r}{C_p} - f \right) \frac{V^2}{V_a^2} \right]} = \frac{I}{d} dV$$

$$ds = \frac{V\,dV}{g \left[\frac{T}{Q} + f + \left(\frac{C_r}{C_p} - f \right) \frac{V^2}{V_a^2} \right]} = \frac{V}{d} dV$$

where $d = \frac{Rg}{Q} = \frac{T + R_a + R_s}{m} =$

$$= g \left[\frac{T}{Q} + f + \left(\frac{C_r}{C_p} - f \right) \frac{V^2}{V_a^2} \right]$$

For the purpose of graphical integration the ratios I/d and V/d are computed and charted as functions of the speed V, and the length of the landing run is then obtained by graphical integration of the curve V/d within the limits $V = 0$ and $V = V_a$; while analagous integration of the curve I/d within identical limits yields the time of the landing run.

If the braking thrust T equals zero, the equation for ds can be integrated analytically, and Glauert's well known relation obtains, which reads :

$$\frac{2 . g . s}{V_a^2} = \frac{\log f/\mu}{f - \mu}, \text{ where } \mu = \frac{C_r}{C_p}$$

This permits to compute the length s of the landing run. We make by way of example $Q = 45{,}000$ kg, $S = 190$ sq. m., $C_p = 0{\cdot}955$, $C_r = 0{\cdot}099$, $V_a = 160$ km per hr. $= 44{\cdot}5$ m per second, assuming the three wheels to be in contact with the ground.

In carrying out the computation it must be considered that a certain length of time, say, 2 seconds, is required to reverse the pitch of the airscrew. It will be assumed that the maximum braking thrust is coincident with a plane speed of 150 km per hour.

LANDING RUN WITH FRICTION BRAKES ONLY.

We assume a ground friction coefficient of 0·15 as compared with a theoretical coefficient of 0·3, taking into account that under normal conditions the brakes are successively applied and released in order to preclude

excessive heating of the brakes and damage to the tyres. With these numerical factors we obtain by the graphical method :

V km/hr	I/d	V/d
160	0·99	44
128	0·835	29·8
96	0·765	20·5
64	0·72	12·8
32	0·697	6·2
0	0·685	0

Length of landing run : 812 m.
Time of landing run : 34·25 sec.

LANDING RUN WITH BRAKING BY TWO AIRSCREWS, BUT WITHOUT FRICTION BRAKING.

On the basis of the assumptions made above, we obtain :

V km/hr.	I/d	V/d
160	0·99	44
150	0·446	18·7
128	0·513	18·3
96	0·643	17·15
64	0·78	13·9
32	0·925	8·22
0	1·01	0

Lenght of landing run for $f = 0{\cdot}03$: 629 m
Time of landing run : 31·7 sec.

If the landing run is executed with 4 airscrews employed for braking, with the friction brakes remaining inoperative, a length of landing run of 441 m and a time of 20·3 sec. are found. If the braking action of two airscrews is combined with the employment of the friction brakes ($f = 0{\cdot}15$), the length of the landing run is obtained as 455 m. and the time as 20·27 sec. If the braking action of 4 airscrews is employed in combination with the use of the friction brakes, the length of the landing run is found as 340 m. and its duration as 14·85 sec. By comparing these values with actual test data as established in the United States, the following table is obtained :

	Actual results m	per cent	Theoretical results m	per cent
Friction brakes alone	824	100	812	100
2 airscrews braking, without friction brakes.	625	76	629	77·5
4 airscrews braking, without friction brakes.	436	53	441	54
2 airscrews braking with friction brakes.	427	52	455	56
4 airscrews braking with friction brakes	363	44	340	42

This table shows that the employment of airscrew braking by reversing the airscrew pitch results in a substantial shortening of the landing run, even if friction braking is not resorted to ; wastage of tyres and maintenance of the friction brakes are therefore reduced. For planes equipped with tail skid this method of braking is of especial advantage, and this applies no less to cases where the length of the landing run is restricted, as, for instance, on the landing deck of an aircraft carrier.

RUSSIA

INDUSTRIAL DEVELOPMENT IN THE URALS.

By G. Vorobyov.

The Chkalov region (formerly Orenburg province) which occupies a large part of the southern Urals and the Caspian lowland, was a predominantly agricultural region in the past. Prospecting conducted during prewar years brought to light deposits of gold, oil, coal, shale, nickel, copper, iron-ore and other minerals in this region. Realization of these plans has completely changed the economic aspect of the region where large industrial enterprises have been built. Extraction of oil commenced eight years ago near Buguruslan in the north-western part of the Chkalov region. Since then the Buguruslan Oil Trust has advanced to a leading place in the " Second Baku," i.e., the system of oil fields situated between the Volga and the Urals. During the last five years this trust has effected a ninefold increase in the production of oil.

Seven years ago the copper and sulphur works at Mednogorsk and the nickel works at Orsk which receive ore from the rich Bliavinsk deposits were put into operation. The Chkalov region has also developed into a machine-building centre. A modern meat-packing plant and an oil refinery which receives oil from Guryev on the Caspian shore along a 500 mile pipeline have been built in Orsk.

Maps published eight years ago do not show the new industrial centres of Mednogorsk and Novotroitsk which now stand in the former barren steppe districts.

The significance and importance of the region's industry increased during the war. 53 new enterprises have been built during the last five years and the number of workers occupied in industry has increased 2·2 times. The region's industrial output in 1944 was 32 times the total volume of the production in Orenburg province in 1913. The new Plan for 1946 to 1950 provides for the further development of the iron, steel and non-ferrous metals industries ; also of machine-building, oil production, etc., in this region.

The southern Urals heavy machine-building plant whose construction is to be completed in 1947 will produce equipment for open hearth and blast furnaces, rolling mills, overhead cranes and blooming mills. This machine-building colossus is already supplying rollers for blooming mills and cooling plates for blast furnaces.

The Orskkhalilov iron steel mills now under construction will use iron ore discovered near Orsk fifteen years ago. This ore contains a large proportion of chromium and nickel. The steel mills will produce natural alloys of high grade pig-iron and rolled metal. It will have four coking batteries, as many blast furnaces, seven openhearths, three bessemer converters, a blooming mill, a powerful rolling mill and a power station. Their production will equal the total output of the iron and steel industry of the Urals during the Tsarist period. A large reservoir and a 30,000 kilowatt power station are being built on the Ural river to secure an adequate water supply to the Orskkhalilov mills.

Considerable investments will be made into the oil-refining industry whose output is scheduled to increase several-fold by the end of the five year period. It is planned to reconstruct and double the size of the machine tool plant in Chkalov and to build two plants for the production of forge shop equipment and fixtures.

The Kuibyshev machine-building plant was built in Buzuluk between Chkalov and Kuibyshev (formerly Samara) during the war. This plant will be expanded during the coming five years when it will add a forge-shop, foundry and assembly shops. Buzuluk will also receive a new plant which will produce machinery for Ministry of Food industry.

Considerable funds will be spent on the reconstruction and extension of the Orenburg railway. New lines will be built from the station of Sara eight kilometres to the west of Orsk to Magnitogorsk and from Chkalov to Ishimbai, en route to Ufa, capital of Bashkiria. Double tracking will be laid on a considerable section to the east of Orsk and the Kuibyshev-Magnitogorsk railway will cross the Chkalov region which will be converted into one of the most important industrial regions of the the U.S.S.R.

CALCULATION OF THE PERFORMANCE CHARACTERISTICS OF PLAIN BEARINGS.

By A. FRÄNKEL. (From *Mitteilungen aus dem Institut für Thermodynamik und Verbrennungsmotorenbau, Eidgenössische Technische Hochschule Zürich*, No. 4, 1944, 134 pages, 65 illustrations.)

(Concluded from August issue.)

(C). EFFECT OF VARIABLE VISCOSITY.

The Law of Viscosity Variation.

It is well known that the viscosity of the oil in the lubricating film varies considerably. In order to obtain, at least, an approximate insight into this effect, the variation of viscosity will be assumed proportional to the variation in oil film thickness. This course has been proposed by Vogelpohl for the cylindrical bearing. Duffing used it for the plane pad. Axial variations of viscosity will be neglected in the case of bearings of finite width.

Fig. 22, based on experiments by Rumpf, shows that the film thickness curve differs much less from the viscosity curve than the mean viscosity (η_b). It is not suggested that the substitution of oil film thickness is in quantative agreement with the actual viscosity variation, but under certain operating conditions (small ψ, large V, medium ϵ values e.g. at high circumferential speeds) it gives smaller deviation from actual values than the assumption of constant viscosity. Hence, based on equation (6) the viscosity equals :

$$\eta = \eta_m (1 + \epsilon \cos \varphi) \quad .. \quad (48)$$

where η_m is a mean viscosity occurring, according to equation (48) at $\varphi = 90$ deg., and which has to be chosen so that deviations are as small as possible. In Fig. 22, for instance, $\eta_m = 0{\cdot}005$ kg.sec.m^{-2} is a satisfactory value, and the dotted curve shows the assumed viscosity curve. This η_m is greatly different from η_b used in Rumpf's calculations. The above method cannot be used for very large values of ϵ as it would result in excessive variations in viscosity.

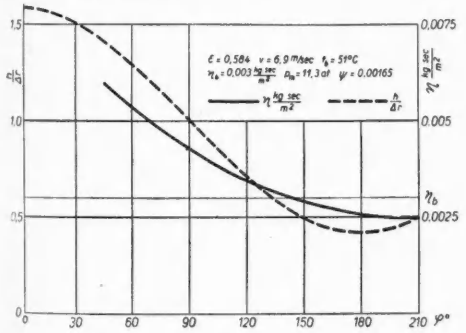


Fig. 22. **Variation of viscosity and oil film thickness in test No. 164 by Rumpf.**

Retaining the other assumptions made in Section A, and the pressure pattern of Fig. 2 ($p = 0$ for $\varphi = 0$ and $\varphi = \pi$), the following equation was developed :

$$p_0 \frac{h^3 m}{3\eta b} + b \frac{V_1 + V_2}{2} \frac{dh}{dx} - \frac{m}{m+1} \frac{b}{12} \frac{d}{dx} \left(\frac{h^3}{\eta} \frac{dp_0}{dx} \right) = 0 \quad (5)$$

This equation is based on a parabola variation of pressure in the axial direction, and holds good for any given bearing width and dependence of η and h upon x. For the bearing at rest $V_1 = 0$; $V_2 = V$. After transformation, introduction of the dimensionless variables Π and Π_0, and substitution of the symbols A and B according to equations (8) and (9) the following expression is obtained :

$$\frac{d^2\Pi_0}{d\varphi^2} - \frac{2\epsilon \sin \varphi}{1+\epsilon\cos\varphi} \frac{d\Pi_0}{d\varphi} - A\Pi_0 = -B \frac{\epsilon \sin \varphi}{(1+\epsilon \cos \varphi)^2} \quad (49)$$

This equation is very similar to equation (10).

In view of the merely orientative aspect of this section on the effect of variable viscosity, the Fourier development of the equation (49) is limited to the special case of $\beta = \infty$. In this case the pressure distribution follows the equation :

$$\Pi_\infty = \frac{p_m \psi^2}{\eta_m \omega} = \frac{6 \epsilon \sin \varphi}{1 + \epsilon \cos \varphi} \quad .. \quad (50)$$

This expression is simpler than equation (35) for η = const. In Figs. 23 and 24 the pressure distribution of an infinitely wide bearing for variable η is compared with that for constant η. Up to $\epsilon = 0{\cdot}6$, the two curves are fairly close together, but for $\epsilon = 0{\cdot}8$ deviations are considerable.

The position of maximum pressure (h_0, φ_0) is derived from equation (50) for $\frac{d\Pi}{d\varphi} = 0$:

$$\cos \varphi_0 = -\epsilon \; ; \; h_0 = r\psi (1-\epsilon^2) \quad .. \quad (51)$$

The maximum pressure itself :

$$\Pi_{g\infty} = \frac{p_{g\infty} \psi^2}{\eta_m \omega} = \frac{6\epsilon}{\sqrt{1-\epsilon^2}} \quad .. \quad .. \quad (52)$$

In Fig. 25 $\Pi_g = \frac{p_g \psi^2}{\eta_m \omega}$ has been plotted, together with the maximum pressure for η = const. For $\epsilon > 0{\cdot}6$, with η = const., pressures are considerably higher than with variable η. The position of maximum pressure is much further away from the point of minimum film thickness for variable viscosity than for η = const. (see Fig. 26).

The resultant journal load is calculated as :

$$\Pi_{m\infty} = \frac{p_{m\infty} \psi^2}{\eta_m \omega} = \sqrt{\Pi_m \xi^2 + \Pi_m \zeta^2} \quad .. \quad (53)$$

Π_m curves are plotted in Fig. 27 for constant and variable η. Again, agreement is good up to $\epsilon = 0{\cdot}6$. Fig. 28 shows the effect on φ_p plotted against ϵ, φ_p being much smaller for variable viscosity. In Fig. 29, the path of the journal centre is indicated. The effect is very considerable and exceeds the influence of relative bearing width.

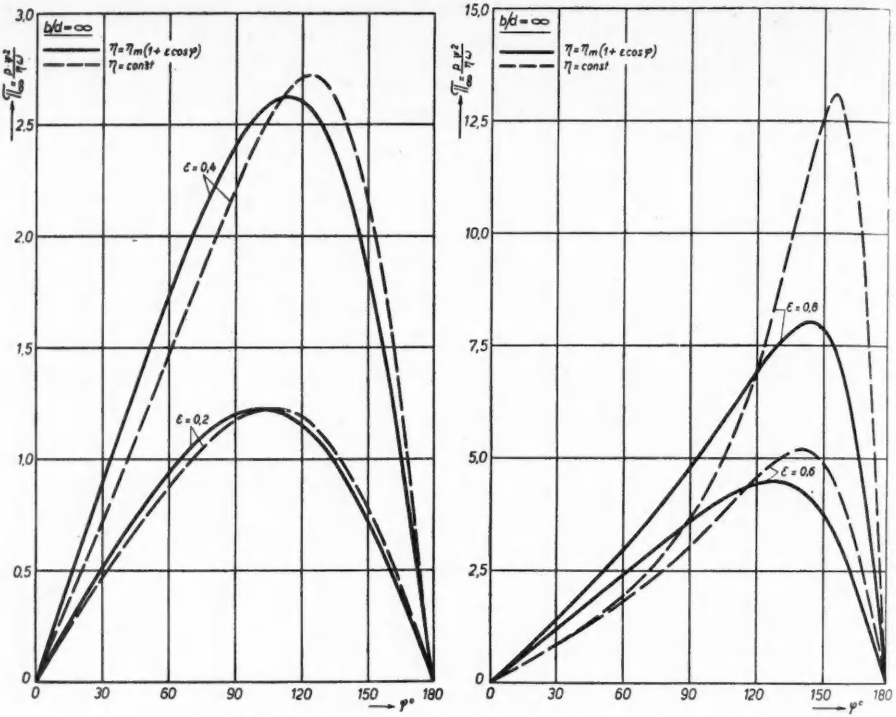
For the bearing of infinite width and variable viscosity the shearing force per unit width and length is developed from equations (2) and (48), leading, after integration to

$$\frac{S\psi}{\eta_m \omega r b} = 4\pi - 3\pi \sqrt{1-\epsilon^2} \quad .. \quad (54)$$

the equation for an ellipse.

$\frac{S\psi}{\eta_m \omega r b}$ is shown in Fig. 30 as a function of ϵ and compared with the curve for η = const. according to equation (40). The shear force for variable viscosity is smaller than for η = const. down to fairly low values of ϵ. In the same figure the coefficient of friction has been plotted for both cases. The differences are quite small.

The main performance characteristics for a bearing of infinite width have now been calculated with assumed viscosity variation. It has been shown that, up to $\epsilon = 0{\cdot}6$



Figs. 23, 24. **Pressure distribution in bearing of infinite width. Viscosity variable.**

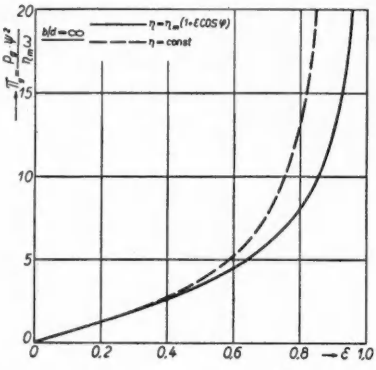


Fig. 25. **Maximum pressure $\Pi_g = \frac{p_g \psi^2}{\eta_m \omega}$ for infinite width and variable viscosity.**

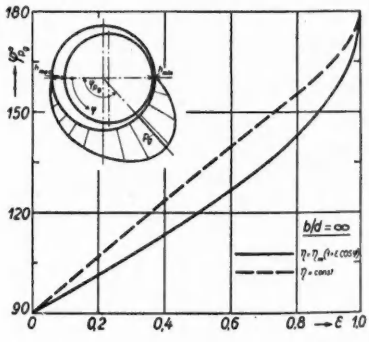


Fig. 26. **Position of maximum pressure, φ_{pg}**

the results obtained for η = const. or $\eta = \eta_m (1 + \epsilon \cos \varphi)$ are more or less identical. For higher values of ϵ, constant viscosity causes overestimating the load carrying capacity and the shear force. Up to about $\epsilon = 0{\cdot}8$ the calculation may be carried out with η = const. provided $\eta = \eta_h$ is carefully chosen. Above $\epsilon = 0{\cdot}8$ the errors tend to become excessive. With relative eccentricities of $\epsilon = 0{\cdot}9$ and more, extreme care is recommended. Up to $\epsilon = 0{\cdot}6$, however, the method of calculation given in section A may safely be used. For bearings of finite width the limitations will be about the same.

The choice of a mean temperature, when calculating with constant viscosity, is very important. The mean between oil inlet and outlet temperature—as used by Nücker—is certainly not suitable. The correct value $\bar{\eta}$ need not necessarily be the arithmetic mean of minimum and maximum viscosity, the definition given for η_m. In view of the importance of pressure distribution and resultant journal load, it can be stated that—for any given law of viscosity variation—the integral

$$\int \Pi d\varphi = \int p \frac{\psi^2}{\eta\omega} d\varphi$$

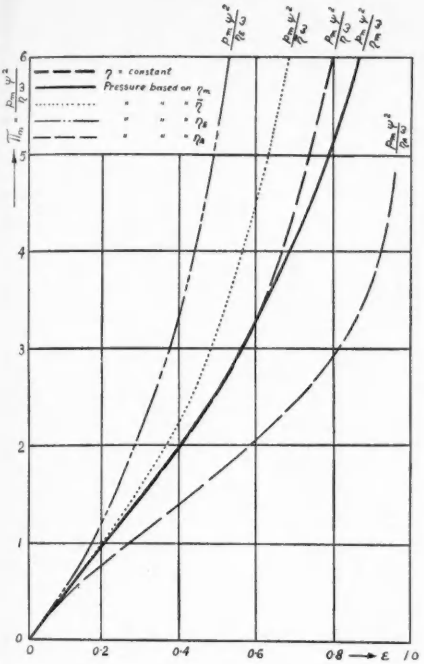


Fig. 27. **Bearing load** $\Pi_m = \frac{p_m \psi^2}{\eta_m \omega}$ **for** $\beta = \infty$ **and** $\eta = \eta_m (1 + \varepsilon \cos \varphi)$.

Compare with results for η = const. The other curves are obtained if the pressure distribution for variable viscosity is based on constant values of η_s other than η_m.

(over the loaded part of the circumference) remains constant, whether η is variable, or $\eta = \bar{\eta}$ = const. From this it follows :

$$\bar{\eta} = \frac{\int p \, d\varphi}{\int \frac{p}{\eta} \, d\varphi} \qquad .. \qquad .. \qquad (55)$$

In general, it will be very difficult to calculate $\bar{\eta}$ numerically from this definition, but expression (55) helps in choosing a suitable mean viscosity. It can be shown, for example, that $\bar{\eta}$ is always smaller than the arithmetic

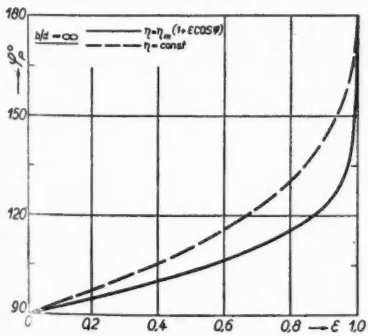


Fig. 28. **Direction of resultant journal load,** φ_p

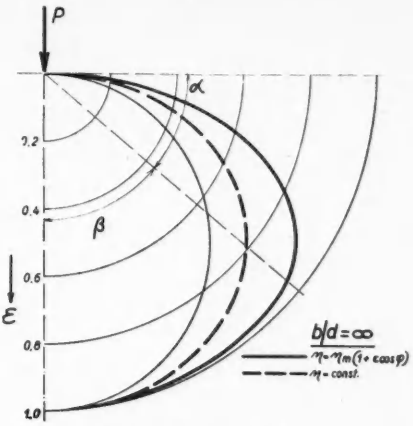


Fig. 29. **Path of journal centre.**

mean η_m. How great the influence of the correct choice of η is shown by the additional curves given in Fig. 27, for η, η_A (initial viscosity) and η_E (final viscosity). Since the deviation of the curve for $\eta = \bar{\eta}$ = const. from the curve for variable η, is greater than that of the curve for $\eta = \eta_m$ = const. it appears that equation (55) requires correction, and in subsequent calculations η_m is preferred.

The Heating of the Oil Film.

In order to check to what extent the assumption of

$$\eta = \eta_m (1 + \epsilon \cos \varphi)$$

is justified, the heat developed in the oil film has been determined. The calculation is again limited to the bearing of infinite width. From the expression for the energy lost owing to friction for a circumferential element $dx = r d\varphi$, in one second :

$$dW = dS \, . \, V — Q dp \text{ mkg/sec.}, \qquad .. \qquad (56)$$

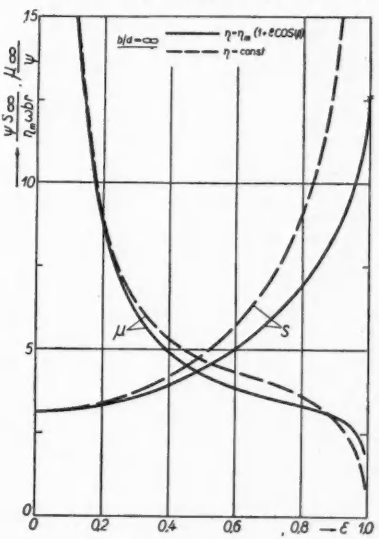


Fig. 30. **Shear force** $\frac{S\psi}{\eta \omega r b}$ **and coefficient of friction** $\frac{\mu}{\psi}$ **in the loaded bearing half for** $\beta = \infty$ **and** $\eta = \eta_m (1 + \epsilon \cos \varphi)$.

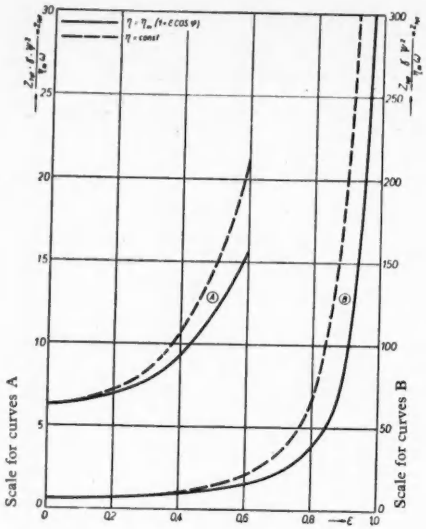


Fig. 31. **Total heat developed in oil film,** $\frac{Z_{tot}\,\gamma\,\psi^2}{\eta_m\,\omega} = z_{total}$, **for $\beta = \infty$ for constant and variable viscosity.**

and from equation (54), (introducing equation (51) and transforming) the following dimensionless expression for the heat developed is derived :

$$\frac{Z\gamma\psi^2}{\eta_m\omega} = \frac{8\varphi}{1-\epsilon^2} - \frac{12}{\sqrt{1-\epsilon^2}} \tan^{-1}\left(\sqrt{\frac{1-\epsilon}{1+\epsilon}}\tan\frac{\varphi}{2}\right) - \frac{6\epsilon\sin\varphi}{1+\epsilon\cos\varphi} \quad .. \quad (57)$$

The total frictional energy per kg. oil is obtained from equation (57) by putting $\varphi = \pi$.

$$\frac{Z_{total}\,\gamma\,\psi^2}{\eta_m\,\omega} = \frac{8\pi}{1-\epsilon^2} - \frac{6\pi}{\sqrt{1-\epsilon^2}} \quad .. \quad .. \quad (58)$$

[for $\eta = \eta_m(1 + \epsilon\cos\varphi)$]

For η = const. :

$$\frac{Z_{total}\,\gamma\,\psi^2}{\eta\,\omega} = \frac{2\pi(1+2\epsilon^2)}{(\sqrt{1-\epsilon^2})^3} \quad .. \quad .. \quad (59)$$

These two expressions are plotted in Fig. 31 showing that when calculating with constant viscosity ($\eta = \eta_m$) considerably higher heat loads are found to obtain than with variable η. Also, with increasing ϵ, the thermal load increases at a much higher rate than the frictional energy. Again it is shown that, for $\epsilon > 0{\cdot}8$, with η = const. very unreliable results are obtained. Besides, it is shown that values of $\epsilon > 0{\cdot}8$ should be avoided for operational reasons because the high thermal load causes excessive heating of the oil film.

So far the present section has indicated the limits within which calculation with constant viscosity is admissible ; the deviations caused by this method of calculation at higher values of ϵ ; and the thermal load on the oil film.

Qualitatively these results also apply to bearings of finite width, especially since the shear force, which is primarily a function of heat generation and viscosity variation in the oil film, is little affected by the bearing width.

The Variation of Viscosity in the Oil Film.

In an approximate calculation of the actual variation of viscosity, the assumption is made that all the frictional heat evolved remains in the oil film. The increase in temperature is calculated as

$$\Delta t = \frac{AZ}{c} \quad .. \quad .. \quad (60)$$

where Z is taken from equation (57). Although the results obtained by this method tend to be lower than those calculated by means of equation (48) it may sometimes be found very useful as a first approximation.

RUSSIA

THE MECHANISM OF THE GYROSCOPE IN CARDANIC SUSPENSION.

By E. L. NICOLAY. (From the 3rd Technical-Scientific Conference of the KALININ—Polytechnic Institute, Leningrad, September, 1944, pp. 16-17).

THE rapidly rotating gyroscope in cardanic suspension is the basic element in the majority of gyroscopic apparatus, such as the gyro-compass, equipment for blind flight, etc. Consequently, the study of the mechanism of the gyroscope is of considerable interest. The author has examined the effect of the masses of the gimbals, and the effect of friction of the cardanic trunnions.

The influence of the masses of the gimbals on the motion of a gyroscope is revealed when investigating this motion under the influence of inertia. This method leads to hyperelliptic quadratures. Special interest is focussed on the influence of the masses of the gimbals on the stability of the axis of a rapidly rotating gyroscope. Although the axis of such a gyroscope is stable in all its positions, the degree of stability of the axis of the gyroscope is considerably reduced when its position coincides with the axis about which the outer gimbal rotates.

The study of the effect of friction at the trunnions of the gimbals on the motion of a rapidly rotating gyroscope is made difficult by the necessity of subdividing the whole motion into separate phases and of investigating every phase separately. This difficulty can be overcome by means of the " image point method."

The friction at the trunnions of the suspension may cause undamped nutating oscillations of the gyroscope axis. Under the action of a constant outside turning moment on the gyroscope the nutating oscillations of the gyroscope axis which accompany its precision are at first damped under the influence of friction at the cardanic supports. However, this damping continues only up to a certain limit beyond which undamped nutating oscillations of the gyroscope axis are produced. An experimental method of determining the frictional torque at the gimbal trunnions from observations of the motion of the gyroscope axis has been evolved.

THE INFLUENCE OF FOUNDRY TECHNIQUE ON THE MALLEABILITY OF METALS AND ALLOYS.

By P. BASTIEN. (From *Fonderie*, No. 3, March, 1946, pp. 99-109, 20 illustrations.)

THE prime material for hot working processes on metals and alloys usually consists of ingots, slabs, or billets cast from the liquid state. The foundry process which precedes the forging, rolling, or extrusion of the material and which influences its malleability has therefore a pronounced bearing on the success of the hot working operation, especially its first phase.

"Malleability" can be defined as the capacity of a material for plastic deformation into a given shape without any internal or external defects. Malleability is a complex property and its different components can be grouped under the following headings :—

Material. Ease of crystal slip, crystal twinning, work-hardening and re-crystallization, inclination to chemical change (oxidation) at forging temperatures.

Work piece. Shape and mass influencing the cooling rate, differential cooling and re-crystallization under internal heat during hot work. The grain structure.

Working process. Different dies and rolls, extrusion from closed container, etc. Speed and extent of deformation.

Contact between die and work piece. Rate of cooling, friction factor on rolling, etc.

No test method is yet available which would allow to discern the separate influence of any one of these factors. The following malleability tests, however, permit to draw general conclusions about the malleability of a material and, by elimination of some variables, to study the separate influence of others.

MALLEABILITY TESTS.

In current industrial practice, direct forging or rolling sampling tests are taken. This method is tedious, costly, and subject to errors and outside influences (e.g., temperature measurements). It could well be replaced or assisted by laboratory specimen tests conducted with the same type of specimen and of deformation, and supplying information, for a given alloy, about optimum temperature range for hot work, extent of deformation obtainable in one heat, and if possible, about the influence of deformation speed. These tests should be simple, cheap, and give reliable reproduceable results.

Tension, compression, bending, and torsion tests have been considered but research on the plastic deformation of both steels (1)* and light alloys (2), has shown the static and impact hot bending tests to be the most suitable for an overall appraisal of the malleability of a new alloy and the determination of maximum and minimum temperatures for hot work. These tests are simple to take, once the apparatus has been built and their results agree well with industrial experience. The impact test can best be used for investigating the influence of different variables, e.g., grain size, impurities on the malleability.

* Figures in brackets refer to the bibliography at the end of this article.

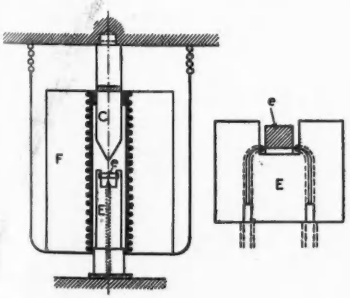


Fig. 1. **Apparatus for static hot bending tests.**

Apparatus for static hot bending tests. (Fig. 1.)

The notched "Mesnager" type specimen e, its support E, and the bending tool C with rounded point are arranged inside an electric furnace F and mounted on a tensile testing machine working in compression. For tests on steel, the furnace is sealed against atmosphere and heated under a controlled atmosphere of pure nitrogen or nitrogen saturated with methyl alcohol vapour.

When the specimen is deformed, the first open cracks will show by an anomaly in the stress-strain record and the observer measures the load at this point. If no fracture appears, the test is continued until the specimen leaves the supports and the load is measured at which plastic flow begins. In both cases, the deformed specimen is examined.

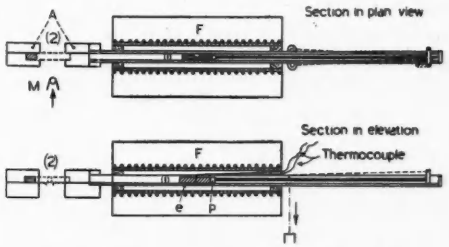


Fig. 2. **Apparatus for impact hot bending tests.**

Apparatus for impact hot bending tests. (Fig. 2.)

The notched specimen e is heated inside a tubular automatically controlled electric furnace F in position 1 and, after reaching thermal equilibirium, is pushed by rod p into position 2 on the supports A of an "Amsler" type impact hammer of 2 kg. m. (173 in. lbs.) impact energy. The pendulum M is automatically released when the specimen is in position so that a minimum time elapses between heating and testing.

If the specimen is fractured, the observer measures the bending angle and energy absorbed. If no fracture occurs, the specimen is bent to a definite angle and then slips in between the supports. The energy absorbed is measured and the appearance of the specimen examined.

In another apparatus for tests on steel (1), the specimen rests on supports inside a vertical electric furnace and is hit by a weight of special shape falling from a controlled adjustable height. Shims of different thicknesses limit the deflection of the test piece and allow to determine, for varying temperatures and deformation speeds, the deflection which just leads to the appearance of the first cracks. If the deflection is thus kept constant, the optimum temperature range for hot work can be established. If the temperature is kept constant, the extent of deformation obtainable in one heat can be ascertained.

FOUNDRY FACTORS INFLUENCING THE MALLEABILITY.

Pure Metals.

The plastic working is influenced by the possibility of crystal slip depending on the lattice structure. Opportunities for slip are best for face centred cubic crystal lattices (Al, Cu, γ-Fe), less good for body centred cubic (α-Fe, Cr, W, Mo), and least for hexagonal lattices (Mg, Zn). Opportunity for slip varies with temperatures, e.g., magnesium offers one slip plane parallel to hexagon base, below 225 deg. C. but twelve more potential slip planes above 225 deg. C. Magnesium alloys should therefore be worked above a certain critical temperature.

Crystal twinning favours plastic working. Face centred cubic lattices (aluminium excepted) incline to twinning, body centred cubic lattices do not. Hexagonal structures form twin crystals if compressed parallel to the hexagon base or stretched parallel to the hexagon axis. Thus, magnesium is easily compressible in one direction but difficult to stretch in the same direction (3) and deformation of magnesium alloy pieces is easier under alternate compression and tension in the same direction.

Alloys.

With a test piece of fine equi-axed grain and homogeneous structure, free of flaws, the malleability of a solid solution light alloy (Al-Cu, Al-Mg, Mg-Cu, or Mg-Al in solid solution) depends on the malleability of the metals in solution, e.g., solid solutions of Al-Cu are better malleable than those of Al-Mg. However, under identical conditions, the load needed to obtain plastic flow is higher for a solid solution alloy than for the pure base metal of this alloy. For Al-Cu and Al-Mg solid solution alloys, this law has been confirmed by the malleability tests as well as by shop experience.

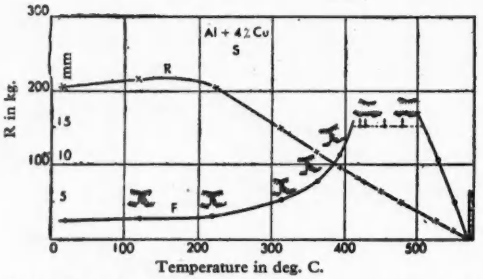


Fig. 3. **Static hot bending tests on sand cast Al plus 4 per cent Cu-alloy (coarse grained).**
R = Load, F = Deflection.

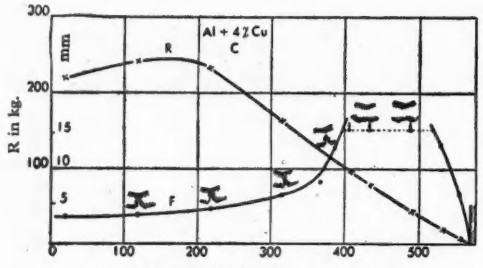


Fig. 4. **Static hot bending tests on chill cast Al plus 4 per cent Cu-alloy (fine grained).**
R = Load, F = Deflection.

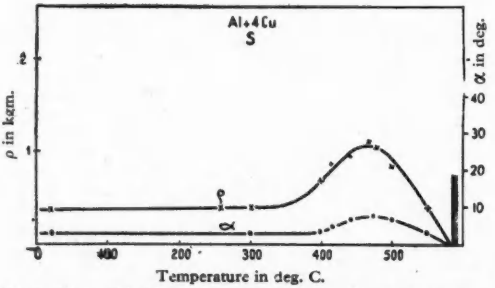


Fig. 5. **Impact hot bending tests on sand cast Al plus 4 per cent Cu-alloy (coarse grained).**
ρ = Energy spent at fracture, α = Angle of fracture.

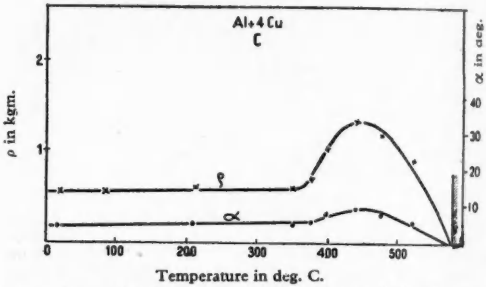


Fig. 6. **Impact hot bending tests on chill cast Al plus 4 per cent Cu-alloy (fine grained).**
ρ = Energy spent at fracture, α = Angle of fracture.

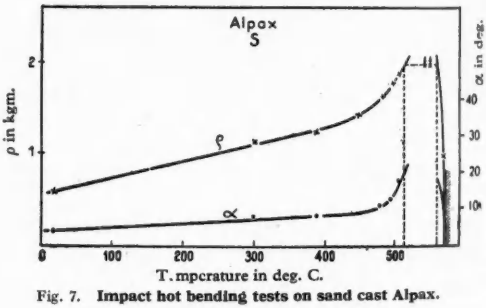


Fig. 7. **Impact hot bending tests on sand cast Alpax.**
ρ = Energy spent at fracture, α = Angle of fracture

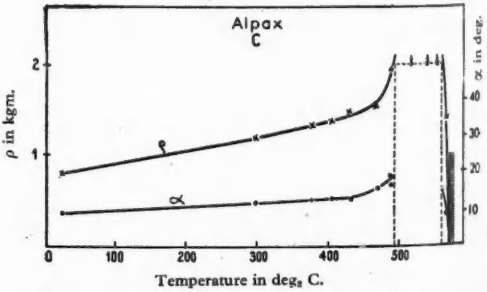


Fig. 8. **Impact hot bending tests on chill cast Alpax.**
ρ = Energy spent at fracture, α = Angle of fracture.

The malleability of two-phase alloys depends mainly on the constituent out of solution, its nature, lamellar or globular form, distribution, and, very likely, its own malleability at working temperature (4). The two-phase alloys Al-Al_2Cu and Mg-Mg_2Cu are satisfactorily malleable but Al-Al_3Mg_2 or Mg-Mg_3Al_2 are notably less forgeable. Two-phase Al-Si alloys contain brittle and hardly malleable eutectic Si-crystals and can only be hot worked within a narrow temperature range (5).

Grain Size.

The grain size depends on the properties of the metal or alloy, on the melt (duration, highest temperature reached, refining or fluxing agent), on the method of pouring and solidification (sand or chill mould). Coarse grained billets must be handled carefully, fine grained material is less sensitive for hot work.

Malleability tests at different temperatures and on a wide range of chill and sand cast light alloys show that the temperature range for hot working a given alloy is the more extended the more the primary grain structure has been refined by chill action or refining agents. (See Figs. 3 to 6.) Tests on Alpax show a narrow but distinctive temperature range within which Alpax is highly malleable, 560 deg. C. to 495 deg. C. approx. for sand cast and 560 deg. C. to 450 deg. C. approx. for chill cast material. (See figs. 7 and 8.) With the necessary precautions, Alpax should therefore be workable, and it has in fact been extruded industrially at 500 deg. C. to 450 deg. C. approx.

Grain Form.

Coarse columnar crystals reduce the malleability of a material and their structure should first be broken down. Moreover, sharp ingot corners may lead to planes of special weakness at the intersection of differently orientated columnar structures and thus cause cracks during the first phase of hot work. (See Fig. 9.) Mold corners should be rounded.

Certain non-oxidizing steels are specially susceptible to a reduction of their malleability by columnar crystal structures, and malleability tests on austenic steels show weakness of intercrystalline cohesion when the plane of fracture lies parallel to the direction of columnar growth (1). (See Fig. 10.) Note that the smallness of the test specimen allows to study the influence of localized and orientated grain structures.

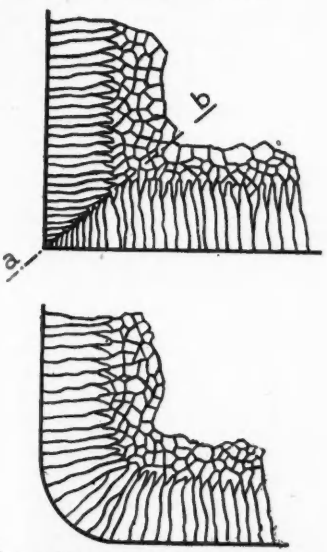


Fig. 9. **Influence of ingot mould design on ingot grain structure.**

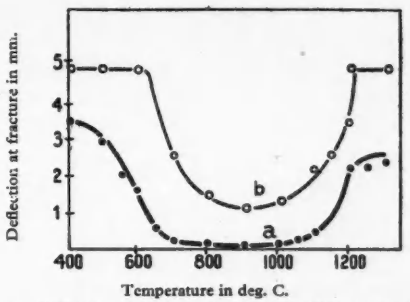


Fig. 10. **Influence of grain orientation in the test specimen (columnar crystal structure) on the malleability of austenitic steel with 0·1 per cent C. and 40 per cent Ni.**

(*a*) Fracture plane of test specimen parallel to direction of dendritic growth.

(*b*) Fracture plane of test specimen perpendicular to direction of dendritic growth.

Segregations.

Segregations accompany the crystallization of alloys on freezing and cause zones of different chemical composition within the single dendrite as well as throughout the whole ingot. They may be zones of weakness and reduce the malleability or lead to internal fracture. They are difficult to analyse quantitatively but generally are the worse the bigger the primary ingot or slab. A homogenizing annealing process at high temperatures for a prolonged time is used to reduce segregations by diffusion, but is often inefficient, notably with steel. Examples of detrimental segregations are herring-bone segreations in duralumin forgings for propeller blades and " stratifications " in certain nickel-chromium armour plating.

Inclusions.

The influence of inclusions depends on their physico-chemical state in the metal at deformation temperature. If in solid solution with the base metal, they change its plasticity and increase the pressure needed for a given deformation at a given speed. If out of solution, the form, distribution, and melting point of the constituent formed by inclusion and the base metal determine its influence on the malleability. Intergranular impurities with a low melting point can prevent any successful hot working of the metal. A small content of bismuth or calcium prevents rolling or drawing of copper.

The importance of inclusions is shown by impact hot bending tests on the following three light alloys: (see Figs. 11 and 12).

(I) Mg = 5·3 per cent.; Mn = 0·59 per cent.; Si = 0·07 per cent.; Fe = 0·11 per cent; rest = Al of 99·8 per cent. purity.

(II) Mg = 6·2 per cent.; Mn = 0·58 per cent.; Si = 0·15 per cent.; Fe = 0·14 per cent.; rest = Al of 99·6 per cent. purity.

(III) Mg = 6·15 per cent.; Mn = 0·51 per cent.; Si = 0·35 per cent.; Fe = 0·14 per cent.; rest = aluminium (75 per cent. of 99·6 per cent. purity, 25 per cent. aluminium scrap).

Alloy III prepared with 25 per cent. scrap can only be hot worked within a narrow temperature range at approx. 380 deg. C.; it is brittle and not forgeable outside this range. The difference in behaviour between alloys II and III cannot be due to the difference in Si-content only, the unspecified impurities like alumina and gas inclusions also greatly contribute to the lack in malleability. The tests are confirmed by industrial experience: Alloy I can be rolled in the normal manner, alloy III cannot be hot worked at all. The impact malleability

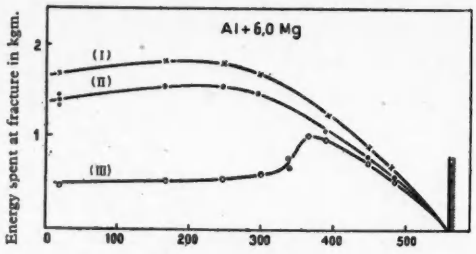


Fig. 11. **Impact hot bending tests on three alloys Al plus 6 per cent Mg, of different purity, energy spent at fracture as a function of temperature.**

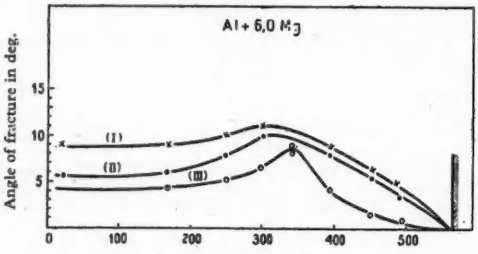


Fig. 12. **Impact hot bending tests on three alloys Al plus 6 per cent Mg. of different purity, angle of fracture as a function of temperature.**

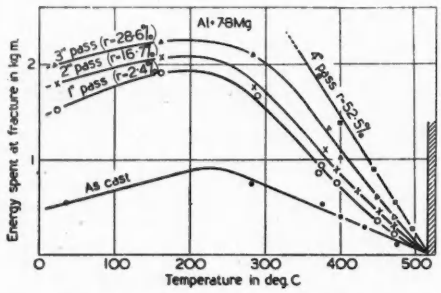


Fig. 13. **Impact hot bending tests showing change of malleability with amount of pre-rolling, energy spent at fracture as a function of temperature.**

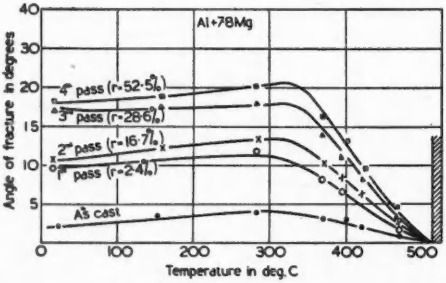


Fig. 14. **Impact hot bending tests showing change of malleability with amount of pre-rolling, angle of fracture as a function of temperature.**

test therefore indicates in a quick and economical way whether a light alloy is forgeable and at what temperature.

Shrinkage and Gas Formation.

These lead to physical discontinuities in the cast material and affect its malleability. Their form, distribution in the ingot and the state of oxidization of the cavity walls are important. Primary pipes would cause cavities or laminations in the forged piece and have to be removed by cutting the ingot top. Secondary pipes and micro-cracks can be closed up and re-welded by hot work. Seams, rokes, and shells often break through the ingot surface, and as their walls are oxidized hot work would only enlarge them. They should be removed wherever possible.

Gas formation leads to blow-holes, their influence cannot, however, be generalized. Small well distributed blow-holes can be re-welded by intense hot work. Globular blow-holes are not detrimental to hot work as shown with rimmed steel where controlled gas formation counteracts the shrinkage and prevents piping.

These foundry defects equally affect top poured and bottom poured ingots but the foundry engineer can try to avoid or displace the defects by varying melting conditions, pouring temperature and speed, cooling power of ingot moulds, or by using hot tops.

Importance of the First Phase of Hot Work.

Coarse grain, chemical heterogeneities, physical discontinuities in the ingot make the first phase of hot work the most critical. Impact malleability tests were taken on specimens from one slab of Al plus 7·8 per cent. Mg, first in its "as cast" state, then progressively rolled to smaller thicknesses at temperatures between 380 deg. C. and 300 deg. C. (see Figs. 13 and 14). The absorbed energy and the angle of rupture increase first quickly, then more slowly, with the amount of hot work done on the test piece before testing. The increase is most pronounced between the "as cast" and the slightly reduced material (2·4 per cent.). The temperature range for rolling the material widens out accordingly. The first slight deformation breaks up the original cast structure and is decisive for the success of the hot working operations, subsequent steps are less delicate and can be bigger.

An alloy of very poor malleability can sometimes be rolled after the cast slab has first been slightly and slowly squeezed under a press. Similarly, cast magnesium alloy slabs are sometimes slowly extruded from a closed container before normal hot work is started.

CONCLUSIONS.

The malleability tests described in this article allow to draw general conclusions about the malleability of a metal or alloy, and to abbreviate the costly preparatory work in rolling and forging mills. Moreover, the tests permit of studying the influence of various factors on the malleability and the results illustrate the importance of the foundry process for successful hot work of a material.

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CLASSIFIED ABSTRACTS

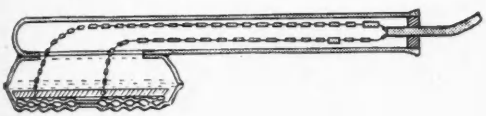
Numerous important articles which, for lack of space, cannot be published in full in our main editorial columns, are regularly abstracted in this section. Subscribers may obtain photostat copies of all original articles at cost.

CHEMICAL ENGINEERING.

Novel Apparatus for Evaporating and Concentrating Liquids by Radiant Heat.

(From *Électricité, France*, Vol. 30, No. 117, June, 1946, p. 132, 3 illustrations.)

A NOVEL apparatus for carrying out evaporation by means of radiant heat embodies as working principle the radiation of heat from above upon the surface of the liquid. The device, which is shown in section in the accompanying engraving, consists of a heating element of the resistor type, in the shape of a coil of nickel-chromium wire. This coil is encased in the appropriately arranged interior grooves of a cast casing of pure silica, the joints of the casing being hermetically sealed by fusing. The heating wires are in contact with the internal side of the radiant surface and they are backed by thermal insulation. The casing is fused to a tube of the same material, which serves as holder of the heater and contains the rubber insulated leads of the heating element.



The various designs of the device are exclusively intended for laboratory use, but it would seem that their use could also be extended to industrial applications. The heating effect is applied solely to the boiling surface and not to the mass of the liquid. Evaporation therefore proceeds with great rapidity on the surface, while the bulk of the liquid remains almost as cold as the container.

Since there is no ebullition, no sputtering of liquid can take place ; while the dry residue remains unaffected by the heating process. In order to improve heating efficiency and to accelerate the heating process, the device may be placed very close to the liquid surface. The silica remains unaffected by acid vapours. In practice the liquid to be evaporated is contained in a dish, the diameter of which is slightly greater than that of the radiant surface, the latter being placed at a distance of 2 to 5 cm. from the surface of the liquid. It would appear that, in order to increase the speed of evaporation still further wherever this is required, heating could be carried out in a partial vacuum.

ELECTROPLATING.

Research concerning Electrolytical Deposition of Metals.

By R. PIONTELLI, M. RUNZA, and M. MINOJA. (From *La Metallurgia Italiana*, Vol. 38, No. 1-3, January-March, 1946, pp. 15-22, 12 illustrations.)

THE authors describe an experimental investigation which was chiefly made for the purpose of studying the initial phase of the formation of electrolytically produced deposits. A general characteristic was found to be the tendency of superficial striations to favour the separation of crystal nuclei. Particularly in the case of low current densities, these nuclei were frequently found to be localised and lined up along such striations. This effect was expecially pronounced with surface layers of aluminium oxide in cases where striations interrupt the continuity of the oxide strata.

This phenomenon was also exemplified by the different behaviour of copper surfaces whether they were mechanically polished or etched with nitric acid ; the latter type of surface was found to reduce substantially the speed of formation of electrolytically produced lead nuclei. These differences in surface behaviour are explained on the basis of the assumption that a polished metal surface closely approaches the amorphous condition, while etching transforms it into a crystalline structure. But in the case of a brass with 70 per cent copper, the difference in the behaviour of a polished surface as compared to that of a surface etched with HNO_3 is less pronounced.

Another interesting result was that an oxidised aluminium surface when used as cathode in the electrodeposition of lead, causes the localised separation of very few nuclei. This is a characteristic which has been intuitively made use of for many years by employing aluminium as cathode material in cases where the deposited metal must be easily removable, as, for instance, in the electrolysis of zinc.

FOUNDRY PRACTICE

Experimental Melting of Cast Iron in Double Shaft Cupolas.

By A. KOCH AND E. O. LISSELL. (From *Gjuteriet, Sweden*, Vol. 36, No. 2, February, 1946, pp. 22-30, 6 illustrations).

DURING the war experiments have been conducted in Sweden to melt cast iron continuously with domestic fuels. The authors describe experiments with a double-shaft furnace fired with charcoal. The intention was to separate fuel and iron in different shafts to avoid crushing of the brittle charcoal and the formation of excess carbonmonoxide in the melting shaft. The experiments show that the method cannot be used in practice. The CO-rich gas from the charcoal in the fuel shaft cannot be produced free from coal particles due to the high gas velocities in the shaft. This disturbs the combustion of the gas in the burner between the fuel and the melting shafts. In spite of the high gas velocity in the coal bed, it is not possible to obtain any appreciable superheating of the iron when passing through the bed. This is obviously due to the reactivity of the charcoal which proved to be too high. Finally it is not possible to melt down the iron in the melting shaft without putting a small bed of coke below the iron column. Other refractory materials have been tried in the bed without success.

INDUSTRIAL WATER TREATMENT.

New Possibilities for Improving the Settling and Precipitating Processes in Reaction Tanks during the Chemical Treatment of Water.

By J. WÜNSCH. (From *Strojnicky Obzor, Czechoslovakia*, Vol. 25, 1945, No. 1, pp. 13-15, and No. 2, pp. 25-31, 8 illustrations.)

THIS article summarizes the work of Dr. Wesly of the I. G. Farben Industry on silica removal combined with lime softening at elevated temperatures followed by final complete softening by means of ion exchange materials. The author incorporates certain theoretical considerations of his own on the design of reaction tank.

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Industrial Water Treatment—*continued.*

Calculations are given which relate the size of reaction tank and velocities, which allow the settling of the suspended precipitated salts, to the size of particle formed in the initial reactions. But it is pointed out that water currents can be set up which upset the settling of the precipitates.

A number of experiments are described and improved results claimed.

INTERNAL COMBUSTION ENGINES

New Engine . . .

(From *Machine Design, U.S.A.*, Vol. 18, No. 7, July, 1946, pp. 125-128, 6 illustrations.)

THE recently developed Jack & Heintz Precision Industries engine is a six-cylinder, horizontal, opposed air-cooled type which incorporates sliding sleeve valves in contrast to the conventional poppet valves. It consists essentially of several aluminium or magnesium die castings, namely, crankcase, cylinder head, oil pan, accessory cover, and front and rear covers. The two crankcases are identical, each comprising a bank of cylinders and half a crankcase, and are bolted together on the centre line of the crankshaft. Each cylinder assembly consists of a pressure spring, reed-type seals, sealing ring, port ring, cylinder, and locating steel segments in addition to the crankcase and cylinder head castings. The lapped surface of the sealing ring mates with the top surface of the port ring, while the other end bears against the pressure spring. When the cylinder hold-down nuts are tightened, the pressure spring is deflected by the cylinder head, imposing a predetermined static load on the complete assembly which actually floats between the two slide valves.

Three distinct features provide the seal that is required during the explosion cycle in an engine of this design. First, the port ring is split and its original tension permits it to perform like a piston ring for oil control by bearing against both slide valves. During the combustion cycle, the gas pressure forces it against the valves, sealing off any leakage that might occur at the ports. Second, the sealing ring is designed to utilize the gas pressures developed during the combustion cycle to increase the force acting on the surfaces of the cylinder, port ring and sealing ring, thus eliminating any leakage at these surfaces. Third, the interference fit between the four thin steel reeds imbedded in the cylinder head and the inside diameter of the sealing ring prevent any gas leakage around the head. These reeds are installed in such a manner that efficiency of the seal increases with increase in gas pressure.

MACHINE TOOLS.

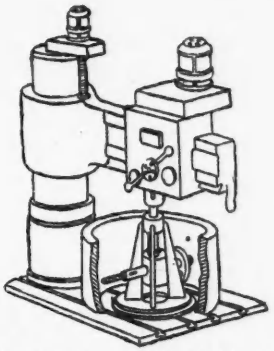
Drilling Machine for Electromotor Frames.

By B. A. EGOROV. (From "*Vestnik Electro-Promyshlennosti*," *Russia*, No. 2, 1946, pp. 23-24, 3 illustrations.)

THE drilling of holes from the outside in the stator frames of medium sized electric motors and generators (say 24 in. to 60 in. int. dia.) is unsatisfactory both because the outer surfaces do not lend themselves readily to accurate marking out since they are being left un-machined, and also because in drilling from the outside, any departure of the drill from its correct direction results in errors at the inside surface, which is the one that matters. Inaccurate location of holes on the inner surface leads to incorrect pole positions, resulting in considerable extra fitting and in a poor quality finished article.

This was eliminated by making an attachment to an ordinary vertical drilling machine to enable it to drill horizontally in all directions as shown in the accompanying engraving.

This device permits of very easy centring of the work by swinging the drill round inside it and using



feelers, so that the holes are truly radial, and the marking out can be accurate since it is made on a machined surface and at the very points where the poles are to be attached. This device is of simple construction and is said to be very effective.

POWDER METALLURGY.

The Use of Ultra-Fine Particles in Powder Metallurgy.

By H. H. HAUSNER. (From *Materials and Methods, U.S.A.*, Vol. 24, No. 1, July, 1946, pp. 98-102, 2 illustrations.)

METALLIC powders when placed in a compacting die will fill the space only to a certain degree and a certain amount of voids will be present. However, if the voids between the coarse powder particles are filled with powders of smaller size, much less deformation of the large particles is necessary to obtain a dense material, and a much lower pressure need be applied for the compacting process. The question is only how fine these particles are to be.

A new type of ultra-fine copper powder and its use for powder metallurgy purposes were tested by the author. The particles of this powder are round, although not quite sperical and of a size between 0·3 to 3 microns. The average particle size is 2 microns, with 90 per cent of the powder said to be of the 2 micron size. To test the usefulness of this ultra-fine copper powder for powder metallurgy purposes, compacts were made with this powder and compared with compacts made from electrolytic copper powder of 200 to 325 mesh size (called "44 to 74 micron powder") and also with mixtures of different particle sizes.

The following four test series were carried out:

Series A:	2 micron powder	100 per cent
Series B:	44 to 74 micron powder	100 „ „
Series C:	2 micron powder	47 „ „
	44 to 74 micron powder	53 „ „
Series D:	325 mesh powder	47 „ „
	44 to 74 micron powder	53 „ „

Four samples of each series were made and compacted at 5, 10, 20 and 30 tons per sq. in. The compacts were then sintered in hydrogen at 1475 deg. F. for one hour and at 1650 deg. F. for one more hour.

Samples made with the 2 micron powder show a low density in the compacted state and a fairly high density in the sintered state.

At low pressure (5 t./sq. in.) the highest density was achieved by using the 2 micron powder.

At high pressure (30 t./sq. in.) the greatest density was achieved by using a mixture of 47 per cent particles of the 2 micron size and 53 per cent of 44 to 74 micron size.

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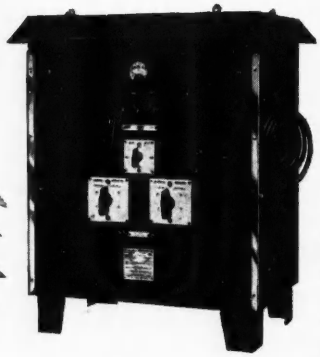

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PUMPS

Rotary-Pump Theory.

By W. E. Wilson. (From *Transactions of the A.S.M.E.*, Vol. 68, No. 4, May, 1946, pp. 371-384, 12 illustrations.)

A THEORY describing the performance of rotary positive-displacement pumps and fluid motors in terms of torque and delivery is presented. From the equations for these two quantities, expressions for power input and power output are developed. The concept of torque efficiency as a complement to the concept of volumetric efficiency is introduced. Equations are developed for the volumetric, torque, and over-all efficiencies. Three operating ranges are identified by outstanding characteristics of the performance in each case. Performance charts in dimensionless form are presented and analysed. Tests on a conventional gear pump operated as a fluid motor yield data which substantiate the theory in one operating range. Additional tests on a cam-type pump give data confirming the predictions of the theory concerning the performance when pumping oils of high viscosity. Alterations of the physical dimensions of the unit are discussed in terms of effect on the performance characteristics. The possibility of the design of units with optimum dimensions is indicated.

STANDARDS AND SPECIFICATIONS.

Standard Classification of Machined Surface Finish.

By P. E. DYACHENKO. (From *Vestnik Inzhenerov i Tekhnikov, Russia*, No. 2, 1946, pp. 73-78, 9 illustrations.)

A U.S.S.R. Standard Classification of micro-smoothness of machined surfaces has been introduced.

It is based on the Root Mean Square value of deviation as measured on an Abbot type surface measuring instrument, and a standard curve of equivalents is given to permit of translating this in terms of maximum deviation should this other criterion be necessary in connection with any particular job.

The classification is into four main Groups. Group I, roughing cuts only. Group II, corresponding to the poorer type of finish encountered in agricultural machinery, etc. Group III, general engineering finish up to and including diamond-turned and polished surfaces. Group IV, extra fine finishes: lapping, honing, super-finish. Each group is to be denoted on drawings, etc., by a corresponding number of inverted triangles.

The groups are subdivided into classes of which there are altogether 14. They are consecutively numbered and each corresponds to a definite range of Root Mean Square values of deviation. Thus: $\nabla\nabla\nabla 7$ denotes the roughest of Group III finishes, Class 7, which corresponds to R.M.S. deviations ranging from 1·6 to 0·8 thousandths of a millimetre.

Still finer differentiation is available by subdivision of classes into sub-grades such as 7a, 7b and 7c, but this is intended for special purposes only, certain industries selecting and standardising a few such sub-grades as required.

The article under review lists the complete classification table, gives the standard conversion curve mentioned above and then proceeds to discuss the degree of roughness resulting from various machining operations, taking into account the form of tool, depth of cut, use of coolants, etc., giving a selection of curves and formulæ, all from Russian sources, together with a summary in the form of a diagram showing the class of finish to be expected from each machining process, depending on whether the operation is intended to be "rough", "finishing" or "fine".

The author then discusses the relative merits and defects and constructional features of various roughness measuring devices and describes the general scheme of introduction of the new system into Soviet industry as follows:—

The Machine Tool Engineering Association is shortly bringing out sets of standard specimens, corresponding to the maximum roughness permitted in each class.

The designer of a new machine part will specify what finish is required.

The works planning department will arrange for routing the part to machines adequate to the task.

The estimator will check visually by comparison with a standard specimen (a magnifying glass with a magnification of at least 1 to 5 is recommended).

Use of surface measuring machines will be resorted to, so far as routine work is concerned, only in the event of doubt or disagreement.

Apart from the Abbot type surface measuring machine the main instruments in use are said to be the Linnik double microscope and the Linnik microinterferometer, both of Russian design and manufacture.

THERMODYNAMICS.

Air Engines.

By H. RINIA and F. K. DU PRÉ. (From *Philips Technical Review, Holland*, Vol. 8, No. 5, May, 1946, pp. 129-136, 5 illustrations.)

RECENT research in Philips laboratories has shown that very satisfactory results can be obtained with air engines, if they are built in accordance with modern conceptions with regard to heat transfer, flow resistance, properties of materials, etc. It was found possible to apply the air process in engines capable of running at 3000 r.p.m., with most satisfactory figures for weight and efficiency.

In this article the theoretical principles of the air process are discussed. It is shown that an installation of two cylinders, in open connection with each other, each closed by a piston with a quantity of air between the two pistons, one cylinder being kept at a constant high temperature and the other at a constant low temperature, acts as an air engine as soon as the pistons are made to move sinusoidally with a constant phase difference; with variations in the volume of the hot cylinder preceding those in the cold cylinder.

It is also possible to reverse the direction of the air cyclic process. It can be made to take place in such a way that upon supplying a mechanical power, a certain amount of heat is taken from a reservoir at a low temperature and a certain amount of heat given off to a reservoir at a higher temperature. This can be accomplished by choosing the phase difference negative.

It is clear that the air process in this way functions as a refrigerator. It has been found in practice that, if the elements corresponding to heater, regenerator, and cooler are given suitable dimensions, very satisfactory refrigerators are obtained.

Approximating Logarithmic Temperature Differences.

By K. D. VOSKRESENSKY. (From *Promishlennaya Energetika, Russia*, No. 2, 1946, p. 11, 1 illustration.)

HEAT transmission in condensers, coolers, etc., is based on the expression:

$Q = K.F.\ \Delta t$ where

K is the coefficient of heat transmission

F area

Δt mean temperature difference

For both parallel and counter-current arrangements Δt has to be calculated from Grashof's formula:—

$$\Delta t = \frac{\Delta t_1 - \Delta t_2}{\ln \frac{\Delta t_1}{\Delta t_2}} \quad .. \quad .. \quad (1)$$

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where Δt_1 and Δt_2 are the temperature differences at inlet and outlet.

As an approximation, it is usual to write :

$$\Delta t = \frac{\Delta t_1 - \Delta t_2}{2} \text{ approx} \quad .. \quad (2)$$

but this is only satisfactory for values of $\frac{\Delta t_2}{\Delta t_1}$ between 0·5 and 1. Beyond that the error exceeds 4 per cent, increasing rapidly.

As many heat transfer devices work outside this limit, there is need for an approximation having a wider range and the author suggests the following equation (3) which gives errors not exceeding 4 per cent for values of $\frac{\Delta t_2}{\Delta t_1}$ between 1 and 0·21 :—

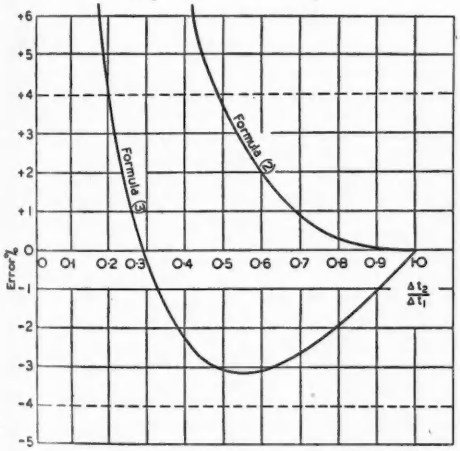
$$\Delta t = \frac{\Delta t_1 + \Delta t_2}{2} - 0{\cdot}1\,(\Delta t_1 - \Delta t_2) \text{ approx} \quad (3)$$

This formula was obtained graphically by plotting Grashof's formula as :—

$$\frac{\frac{\Delta t}{\Delta t_1}}{\ln \frac{\frac{\Delta t_2}{\Delta t_1} - 1}{\frac{\Delta t_2}{\Delta t_1}}} \text{ using } \frac{\Delta t}{\Delta t_1} \text{ and } \frac{\Delta t_2}{\Delta t_1} \text{ as coordinates,}$$

choosing a straight line approximating to it within 4 per cent and having a simple equation, namely

$$\frac{\Delta t}{\Delta t_1} = 0{\cdot}4 + 0{\cdot}6\,\frac{\Delta t_2}{\Delta t_1}$$



and re-writing this in the more convenient form (3) given above.

The errors resulting from the use of the two approximations discussed above are shown in the accompanying illustration.

WELDING.

The New Brown Boveri Automatic Arc Welding Head

By H. Kocher. (From *Brown Boveri Mitteilungen, Switzerland*, Vol. 33, No. 4-5, April-May, 1946, pp. 107-111, 5 illustrations.)

The increased demand for automatic arc welding in repair and fabrication has led to the development of a simple, compact, and rigid automatic welding head for arc welding with bare electrode wire of up to $\frac{1}{16}$ in. diameter. A patented addition also provides for automatic coating of the bare wire as it passes through the welding head, the coating being fed in in the form of powder.

The wire is fed into the arc by means of two rollers driven by a d.c. feed motor which is supplied from a Leonard motor-generator set coupled with an exciter-generator for constant but adjustable excitation of Leonard and feed motors. An impulse reactor coil is arranged to be normally without current but to carry current and react sharply and speedily on the excitation as soon as the arc length, and therefore the arc voltage drop, deviates from its nominal value. The sensitivity of this coil can be adjusted by a regulating resistance and arc length is correctly maintained even over substantial surface irregularities. When striking the arc, the coil actually reverses the feed motor thus lifting the electrode from the parent material until the correct arc distance has been established.

Manipulation of the automatic head is simple and controlled from a switchboard on the head. The feed motor is switched on or off simultaneously with the switching, by relay, of the d.c. or a.c. welding current, and can also be reversed. The nominal arc voltage drop is adjusted within certain limits by a turning knob. Another switch serves a motor slung underneath the head driving an eccentric with electrically regulated speed and manually variable eccentricity of the electrode oscillating movement, thereby imitating the "weaving" of the electrode when depositing a wide bead. The electrode polarity is changed by simple switches on the board and the welding generator. The weight of the head is 150 lbs.; it can be tilted up to 45 deg. in every direction. A special suction funnel can be provided for protection against obnoxious arc fumes.

Underwater Welding.

By C. Linnander. (From *Svetsaren, Sweden*, Vol. 11, No. 1-2, January-March, 1946, pp. 1-9, 12 illustrations.)

In Sweden an underwater cutting torch operated with watergas has been developed by Aga-Faxius which is suitable for underwater depths up to 35 metres. The nozzle piece of the torch is equipped with a special annular nozzle to which compressed air is supplied. The escaping air displaces a sufficient amount of water to permit a relatively undisturbed combustion of the gas mixture.

Electric arc welding under water was tried out in Sweden occasionally during 1941 and on a larger scale in 1944. No fundamental difference between the characteristics of the arc and its generation under water and in air respectively was found to exist. However, since the ionisation of water is more difficult to achieve than that of air, the employment of thickly coated electrodes is required, and higher voltage and amperage must be used. As the electrode and the electrode holder cannot be insulated completely, a loss of current through the water takes place. The magnitude of this loss will depend upon the voltage, which in turn depends upon the salt content of the water, the distance between the exposed terminals, and their surface area.

In sea-water the welding electrode must be given positive polarity. Ordinary direct current welding sets can be employed, but for reasons of safety the no-load voltage must not exceed 75 volts. The electrode holder must be better insulated than is required for operation in air. A 0·2 per cent plain carbon steel with an ultimate tensile strength of 50 kg. per sq. mm. can be welded satisfactorily but the thickness of the workpiece must not be less than 5 mm. The ultimate tensile strength of the underwater weld approximates to that obtained in air.

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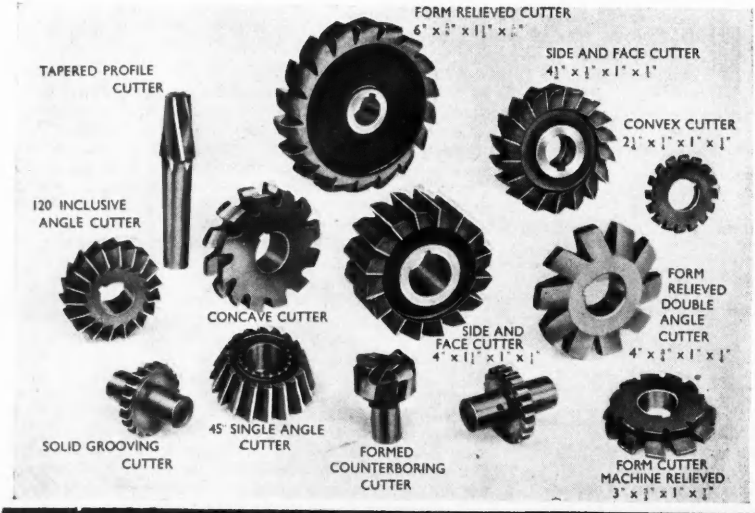
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NEWS OF THE MONTH

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.
Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- PERSONAL
- NOTES

PERSONAL

Mr. T. W. Y. Alderton has been appointed a Director of Pearson and Dorman Long Ltd.

Mr. William Batt, who was Vice-Chairman of the American War Production Board and representative of the United States on the Combined Production and Resources Board, is paying a short visit to this country.

While he is here **Mr. Batt** will address the Institution of Mechanical Engineers and representatives of other British Engineering Institutions, on the progress being made by the United States in the unification of engineering standards between the countries using the inch system of measurement.

Dr. M. L. Becker has taken up an appointment as Superintendent Metallurgist to the British Iron and Steel Research Association.

E. H. Jones (Machine Tools) Ltd., announce that **Mr. A. V. Graves** has joined the Company as Manager of the Used Machine Tools section, with a view to expanding this department. A further development in the expansion policy of the Company is the appointment of **Mr. E. W. Kelleher** as Inside Sales Manager, following **Mr. P. E. Verrall's** vacation of the post to take over the duties of Assistant General Manager.

Sir William Griffiths, D.Sc., F.R.I.C., F.I.P., Chairman and Managing Director, Mond Nickel Co. Ltd., **Mr. H. J. Allcock, M.Sc., M.I.Mech. E., M.I.E.E.,** British Insulated Callender's Cables Ltd., and **Mr. Christopher F. S. Taylor, M.C.,** Rhokana Corporation Ltd., have been appointed to the Management Committee of the Copper Development Association.

Mr. R. J. Hall has been appointed a Director of British Insulated Callender's Cables, Ltd.

Vice-Admiral C. Holland, C.B., who commanded H.M.S. "Ark Royal" during the war has joined A. C. Cossor Ltd.

Mr. W. T. Marshall, B.Sc., A.M. Inst.C.E., technical officer of the Institution of Structural Engineers, has been appointed by the University Court of St. Andrews to the Chair of Engineering and Drawing, University College, Dundee.

Mr. H. R. Mills, M.Sc. (Lond.), A.M.I.E.E., has been appointed Assistant Director of the Science Department of the British Council.

The University of London announces the following appointments:—**Dr. Allan Dawson Ross** to the Chair of Civil Engineering, tenable at King's College; **Dr. Owen Alfred Saunders** to the Chair of Mechanical Engineering, tenable at the Imperial College of Science and Technology; and **Professor Willis Jackson** to the Chair of Electrical Engineering, tenable at the Imperial College of Science and Technology.

Mr. G. N. Smibert has been appointed an additional Director of Richardsons, Westgarth and Co., Ltd.

NEW LITERATURE.

"Nickel Bulletin" Vol. 19, No. 5.—The "Nickel Bulletin" for May, 1946, contains abstracts covering the polarographic analysis of nickel compounds, hard magnetic materials, glass-to-metal seals, the overheating of steel, the creep properties of Monel and Inconel and the uses of nickel and nickel alloy in the pulp and paper industry.

Copies of the "Nickel Bulletin" may be obtained, free of charge, from The Mond Nickel Company, Limited, Grosvenor House, Park Lane, London, W.1.

"Safety Electrical Equipment and Its Use". By S. W. Richards, M.I.Min.E. (Mines & Flameproof Equipment Dept.), The General Electric Co., Ltd.

The second edition of this booklet describes the constructional details of almost every kind of flameproof and intrinsically safe electrical apparatus, and also explains the need for taking stringent electro-mechanical precautions in the manufacture and installation of such equipment. It includes new descriptive matter on the use of fluorescent lighting in mines and other inflammable situations; on the design of G.E.C., flameproof tubular, and immersion heaters, and on the design of a new G.E.C. flameproof conduit joint. Additional notes are given on the characteristics of inflammable gases and liquids. A table shows, at a glance, the flashpoints, ignition temperatures and the explosives limits of the many solvents and gases now used in industry.

The booklet is free to those who are interested, and application should be made to: The Publicity Organization, The General Electric Co., Ltd., Magnet House, London, W.C.2.

New British Standard for Welding Symbols (B.S., No. 1303).—The above specification standardises the symbols used on shipbuilding drawings. The scheme is based on the following:—

(1) That the symbol is in all cases a simple diagrammatic section of the weld metal of the joint.
(2) That the top of the symbol always represents the near side of the weld.

This standard has been prepared as it has been found that a system of welding symbols is a necessity for use with shipyard drawings. There is often considerable exchange of plans between collaborating shipbuilders, the main structural plans are submitted to an approval authority and personnel move from one firm to another so that it is desirable that the system be standard throughout the industry.

An attempt has been made to eliminate, so far as practicable, danger of confusion arising in the interpretation of the symbols. It does not seem practicable to indicate clearly by means of a symbol complete details of a joint, and therefore the present system has been devised for use in conjunction with a key schedule of joints giving complete details of the edge preparation, gaps and welding procedures over the full range of material thicknesses for each type of joint included in the practice of the particular shipyard.

Copies of the specification can be obtained from the British Standards Institution, 28, Victoria Street, London, S.W.1. Price One Shilling each.

DISPERSAL UNITS CLOSED DOWN

H. M. Hobson (Aircraft and Motor) Components Ltd., announces the closing down of its various "dispersal" units and the entire organisation will be located in the group of factories at Wolverhampton. The branch establishment at Park Royal will remain in being. The address of the head office is Hobson Works, Stafford Road, Wolverhampton (telephone, FORdhouses 2266).

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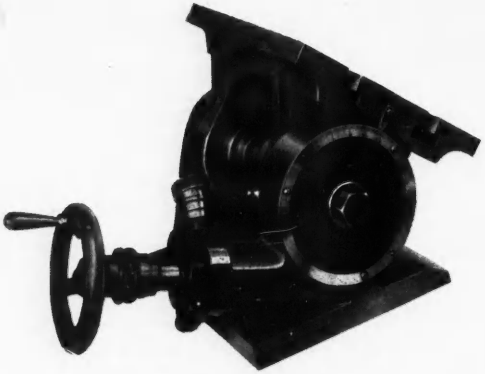

Marlco worm operated angle plate.

NEW EQUIPMENT

Worm Operated Angle Plate.—The "Marlco" worm operated angle plate shown in the illustration is made by W. H. Marley & Co. Ltd., 105, High Road, London, N.11. It is designed for long life and to ensure accurate work. The table base and worm wheel, made of Meehanite, hinge on 2 in. diameter steel trunnions. The operating worm of hardened steel runs in an oil bath situated in the base and can readily be engaged and disengaged with the worm wheel or adjusted to take up any back-lash. At the geared end, the worm wheel has a large diameter cone which contacts a corresponding cone in the body. The wheel drives the trunnion through a sliding feather key and the two faces are adjustable by a nut situated on the trunnion. At the inner end of this trunnion another key engages the table and so unites the worm wheel, trunnion, and table, at the same time allowing uninterrupted adjustment. All working faces are lubricated through easily accessible Tecalemit lubricators. A 5½ in. diameter graduated dial is located at the geared end engraved 0 deg. to 90 deg. clockwise, and 0 deg. to 30 deg. anti-clockwise. In addition, a sleeve behind the handwheel is marked into four main divisions each giving 1 deg. movement of the table. Each of these main divisions is again divided into 12 sub-divisions each giving 5 minutes movement of the table. Rotary adjustment of the sleeve is provided to allow the zero mark to be brought into true alignment if necessary.

A stop with fine adjustment is provided on the base which comes into contact with a hardened steel insert located on the underside of the table when it is 90 deg. from the horizontal. The stop is accurately set and locked before leaving the works. The overall dimensions are :—table 10 in. × 8 in., base 10 in. × 7⅝ in., height 8 in., and the weight is 75 lbs.

New Miniature Lamphouse.—The range of photo-electric accessory equipment made by the General Electric Co. Ltd., has just been augmented by a new miniature lamphouse known as Type M.L.H.

This has been produced to meet the demand for a compact light source suitable for industrial applications where the distance to be spanned does not exceed 3 feet, and where the existing types of amphouse cannot be used.

The lamphouse is designed for use in conjunction with the G.E.C. MD photocell relay amplifier. An Osram 12 volt, 6 watt automobile lamp is used as the light source. To ensure adequate life this should be operated from the 10 volt winding provided on the amplifier transformer.

Telescopic adjustment of the lens enables the light beam to be focussed accurately. The lamphouse is made of brass and is provided with a fixing block drilled and tapped 4 B.A. at ⅝ in. centres. The overall length, including S.B.C. lampholder, is 3¾ ins. and the total height is 1.11/32 ins. The external diameter is 1.1/16 ins.

THE "PHILOWATT"—A NEW 1,000 WATT AMPLIFIER.

A NEW 1,000 watt amplifier designed for very large Public Address installations, or for Wire Broadcast systems serving networks of 6,000 subscribers, has been introduced by Philips Lamps, Ltd.

The "Philowatt," as it is called, is a high quality amplifier with the remarkably low distortion level of 1 per cent. It incorporates many novel features including "S.A.M." (Scott Automatic Monitor) which fulfils a function closely parallel with that of "George" the automatic pilot, in aircraft.

S.A.M. consists of a unit built into the amplifier which provides electronically delayed main H.T. switching, and automatic monitoring of the amplifier output. This facility is provided to allow the unattended operation of the amplifier.

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